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Exhaust gas recirculation study on dual-fuel methane combustion

Master´s thesis

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ABSTRACT OF THE MASTER'S THESIS

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Abstract:

Currently, interest in different alternative transport fuels is growing. There are two main reasons for this motivation: the universal environmental concern, caused by the noticeable impact of petroleum fuels on the human health and environment, and the necessity of replacement due to the declining availability of petroleum.

Dual-fuel engines could mean a partial solution of these worries, since the primary fuel of this type of engines is natural gas, mostly formed by methane, which can derived from both renewable and fossil sources. Diesel consumption is reduced by the introduction of some gas fractions, and dual-fuel mode allows, mainly, fewer greenhouse gas emissions.

From the economic and environmental point of view, dual-fuel is, thus, highly desirable. However, from a technical standpoint, this technique has been proved to be difficult to realize and optimize. Therefore, this master's thesis has the objective to serve as a support on dual-fuel research field, as far as the Exhaust Gas Recirculation (EGR) effect on combustion and emissions performance is concerned.

Emphasis on the thesis was set, thereby, on the EGR rate variation effect on different engine parameters, such as lambda (λ), cylinder pressure, exhaust gas temperature, charge air temperature, heat release rate (HRR), cumulative heat release rate (HR) or ignition delay, as well as on the NO_x, THC and CO emissions performance.

Based on the results of this master's thesis it can be concluded that EGR rate raises lead on significant reduction of NO_x emissions, as well as a slightly decrease of unburned hydrocarbons. CO emissions are not influenced at low EGR rates, performing a sudden increase for the highest ones, around 25-30%. At the same time, combustion efficiency seems to decline with EGR, as seen from HR curves and CO emissions, obtaining extremely rich mixtures.

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Preface

This master´s thesis has been possible thanks to the Energy Technology Department of Aalto University, in particular the research group of Internal Combustion Engines. I would like to express my gratitude for offering me the opportunity to increase my knowledge in the interesting field of internal combustion engines and new alternative fuels.

Of course the work could not have been successful without the cooperation of all the contributors. I owe thanks to my instructor, Lic. Sci. (Tech.) Teemu Sarjovaara, for supervising my work and provides me feedback during the execution of the thesis, as well as Professor Martti Larmi for offering me this project and all the help received. Equally, I wish to thank to my supervisor in Spain, Jesús Casanova Kindelán, for the distance monitoring of my work.

Also some other people from the research group of Internal Combustion Engines have contributed to the project and are gratefully acknowledged, especially Olli Ranta, Otto Blomstedt, Rasmus Pettinen, for all the technical help received, and last but not least, Juhani Törmänen. This thesis would not have been possible without his help and his guidance from my arrival to this research group, so my special thanks belong to him.

I am grateful to all my friends for sharing with me all these years and helping me to get my objective more pleasantly. Without them it would have been extremely arduous to finish my studies.

Lastly, I would like to express my particular thanks to all my family, but especially my parents, Manolo and Reyes, and my sister, Marta, for all the support given during all my life. I am absolutely sure I would not have got to this point without them. Likewise, this special acknowledgement also belongs to Ana López, that special person who is always beside you and knows how to make you stronger at every moment.

Espoo,

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Nomenclature

a:	Crank radius (m)
A/D:	Analog-to-Digital
AFR:	Air-fuel Ratio
B:	Cylinder bore (m)
BSFC:	Break Specific Fuel Consumption (g/kWh)
BTDC:	Before Top Dead Center
°CA:	Crank angle degrees
CARB:	California Air Resources Board
CH_4:	Methane
CI:	Compression - Ignition
CNG:	Compressed Natural Gas
CO:	Carbon monoxide
DOC:	Diesel Oxidation Catalyst
CO_2:	Carbon dioxide
ECU:	Electronic Control Unit / Engine Control Unit
EGR:	Exhaust Gas Recirculation
EHVA:	Electro Hydraulic Valve Actuator
EPA:	Environmental Protection Agency
GER:	Gas Energy Ratio
HC:	Hydrocarbons
HR:	Cumulative Heat Release Rate (J)

HRR: Heat Release Rate (J/°CA)

H_2O: Water

l: Connection road length (m)

LNG: Liquefied Natural Gas

NO_x: Nitrogen Oxides

O_2: Oxygen

PID: Proportional-Integral-Derivative controller

PM: Particulate Matter

PN: Particle Number

ppm: Parts-per-million

Q_n: Net heat release rate (J/°CA)

SCR: Selective Catalyst Reduction

SI: Spark - Ignition

SOI: Start of Injection (°CA BTDC)

SO_x: Sulfur Oxides

SO_2: Sulfur dioxide

STP: Standard Temperature and Pressure

THC: Total unburned hydrocarbons

TWC: Three-Way Catalytic Converter

VOC: Volatile Organic Compound

λ: Lambda / Air-fuel equivalence ratio

γ: Isentropic constant

1. Introduction

Currently, there is a large interest in alternative fuels. Future research work on reciprocating internal combustion engines is highly determined by the progressive depletion of crude oil sources. Moreover, over the recent years, the continuous and growing concern on environmental damage produced by engines has forced the governments to improve their environmental policy, making the fulfilling of the emission standards even more complicated for engine manufacturers.

In order to overcome these concerns, different solutions are investigated at the moment. From the crude oil depletion point of view, numerous methods have been developed, e.g., dual-fuel engines, but a large progress should be made to displace gasoline and diesel at all. On the other hand, some different strategies and after-treatments have been recently included on engines with the objective of reducing the emissions and, thus, fulfil the demanding standards set by administrations. These strategies can be applied by modifying the engine design and its components, such as *Exhaust Gas Recirculation* (EGR), as well as by adding some exhaust gas after-treatment systems, like *Selective Catalyst Reduction* (SCR) or *Three-way Catalytic Converters* (TWC).

For the execution of this master's thesis, a dual-fuel engine with external EGR integrated system has been used. Dual-fuel engine uses typically natural gas, mostly containing methane as main fuel, while a small pilot amount of diesel is used for initiating combustion and igniting the methane, due to the higher reactivity of diesel. This diesel substitution leads to less dependence from crude oil, as natural gas is one of the most abundant fossil fuels at the moment. Moreover, it has been proved that NO_x emissions decrease while increasing the fraction of methane within the dual-fuel mixture. Also other dual-fuel engines have been studied, such as alcohol, typically ethanol, with diesel. However, natural gas – diesel blend is the most common at the moment.

Exhaust Gas Recirculation can be divided into two different methods: internal and external. Internal EGR is obtained from some exhaust gases which remain inside the cylinder after the exhaust stroke, because of the modification of valve timings or camshaft designs. External EGR constitutes the forced recirculation of the exhaust gases into the intake manifold, where they mix with the charge air before entering the cylinder. Second one was used during the experiments in this thesis. EGR influence is mainly NO_x reduction, which happens due to two basic reasons. First, increase of inert gas concentration, which reduces the oxygen concentration available for combustion, and second, the increase of specific heat capacity of the charge. Both lead to lower combustion temperatures that reduce, thereby, NO_x emissions.

The main objective of this master's thesis was to study the EGR rate parameter effect on emissions and engine performance on dual-fuel methane combustion, using a single cylinder high-speed research engine. Different parameters were analyzed for this purpose, such as lambda (λ), cylinder pressure, exhaust gas temperature, charge air temperature, heat release rate (HRR), cumulative heat release rate (HR) or ignition delay. NO_x, THC and CO emissions were examined.

2. Background

As it is known, due to the depletion of the crude oil resources in the world and its probable disappearance in the future, several research works are presently being done in order to find non-petroleum fuels for internal combustion engines. Furthermore, this kind of petroleum fuels causes high levels of exhaust gas emissions, which can be unhealthy for humans and harmful for the environment.

Owing to this universal environmental concern, interest in generating alternative fuels for internal combustion engines has become higher over the last few years. Thus, the main point is to discover alternative sources of fuel energy supply, attending at the same time the needs for social and economic progress, and the difficulty in some rural communities to get fossil fuels.

Gaseous fuels could be a partial solution to this concern. It is becoming more positive for researchers and users in comparison to the past, since natural gas is one of the fossil fuels with less environmental impact. Gas combustion produces insignificant NO_x quantities, one of the main causes of the acid rain, and it also reduces the amount of CO_2 emissions compared to other fuel combustions, which takes a substantial place in the greenhouse effect.

Most of the combustion machines can be adapted, subsequent to several modifications, for using with gaseous fuels in power production. Gaseous fuels have high octane numbers and it is adequate for relatively high compression ratios, because, in principle, they are able to resist knock better than other liquid fuels.

For these reasons, it can be concluded that it is economic and environmentally desirable to use gaseous fuels in diesel engines.

2.1. Natural gas

2.1.1. Introduction

Natural gas is a fossil fuel containing mainly methane, generally above 85%, and small amounts of propane, butane, ethane and inerts (typically nitrogen, carbon dioxide and helium). It is one of the less pollutant alternative fuels, and it can be stored and used as Compressed Natural Gas (CNG) or Liquefied Natural Gas (LNG) [1].

It is considered a promising alternative fuel to diesel. Its use has been growing over the last years because of its substantial economic and environmental benefits. Its low cetane and high octane numbers make it easy to adapt to spark ignition engines, but in compression ignition engines it is not as easy [2].

It is a clean burn fuel, frequently described as the cleanest of all the fossil fuels, and it is available in many parts of the world. Also trucks and other vehicles, with engines specially designed for this fuel, produce, generally, fewer emissions than gasoline or diesel, as well as being cheaper than these ones. Due to its high octane rating, it has the potential to optimize the thermodynamic efficiency through a high compression ratio [1,2].

One of the disadvantages of natural gas in dual-fuel operation is the slightly high production of “methane slip” or unburnt methane, which is a greenhouse gas 21 times worse than CO_2 in terms of global warming. NO_x production is reduced in comparison to diesel but still constitutes a problem if actions are not taken, so that the EGR system helps to keep it in adequate values. In addition, difficulty of transportation between big distances is also a problem, as the tank must be far larger, heavier and more expensive than conventional ones [3].

Natural gas can be used in two different types of engine: Spark-ignition (SI) and Compression-ignition (CI) engines. The first ones run with the injection of gas and its subsequent ignition produced by spark, producing high thermal efficiency, but increasing NO_x emissions too. However, carbon dioxide (CO_2), unburned hydrocarbons (HC) and carbon monoxide (CO) emissions are usually reduced. On the other hand, CI natural gas engines should be used in dual-fuel mode, as it will be in this thesis, where diesel is injected with the gas in order to provide an ignition source. Thermal efficiency is generally maintained in comparison with typical diesel engines, NO_x and CO_2 are usually reduced, and THC and CO emissions increased [3].

2.1.2. Availability and distribution

Availability in Europe is good, counting on a dense pipelines network to distribute natural gas among industrial, commercial and domestic consumers. Apart from domestic production, it is complemented by considerable imports from Algeria and, mainly, Russia [4].

Natural gas reserves around the world are numerous, but European production is set to decline during the next decade. At the same time, the amount of imports in the European supply will continuously increase, and Russia is supposed to be one of the most credible long-term major supply sources for Europe [4].

As well as the rest of gaseous fuels, CNG needs a dedicated infrastructure for storage and distribution. Natural gas network can be used to supply it to refueling stations. Some areas of Europe are not covered by the grid, and it is unlikely that transport demand alone would justify extensive additions to the existing network. It is in this case when LNG could be an appropriate option, which is distributed by road and maritime ways [4].

LNG can be transported with more efficiency. It can be transported by ship for long distances and also via specially-built trucks to supply local distribution systems. Once a LNG ship arrives to the destination, it is typically off-loaded into well-insulated storage tanks in order to be reconverted into its gaseous form [4].

Refueling stations constitute an important part of the infrastructure and costs. Taking into account that existing conventional fuels emplacements are used, investment and operating costs are mainly related to storage, compression and refueling hardware [4].

2.1.3. Natural gas properties

Some physical properties of natural gas that should be taken into consideration, since they can affect its performance in engines, are:

- Natural gas is a fossil fuel formed from plant and animal remains from millions of years ago, constituting a hydrocarbon component with methane as its major component.
- It is colorless and odorless, although a commercial odorant is added during transportation or processing for safety reasons, in order to allow users to detect the gas.
- The specific gravity is lower than air (0.6 - 0.7), as natural gas is lighter. In case of leaking, it disperses upward and vanishes into the air rapidly.
- A positive attribute of natural gas is its high ignition temperature and its narrow flammability. It is inflamed around a range of 5-15% by volume of gas in air, with a self-ignition temperature of 537 - 540 Celsius degrees.
- The density of natural gas under STP (air at 0 ºC and 1 atm) varies between 0.7 and 0.9 kg/m^3 [5].

Either natural gas or methane is considered depending on the studied values. In order to understand better the methane and natural gas performance in an engine, some other factors and properties are analyzed below, such us densities and heating values. Also diesel properties are included in tables of this section, as it provides a most known source to compare with.

The theoretical air-fuel ratio, density and energy density of a stoichiometric mixture, as well as the relative energy density can be seen in table 2.1.

Table 2.1: AFR and energy density of stoichiometric air-fuel mixture [6].

Fuel	AFR	Mixture density (kg/m^3)	Energy density (MJ/m^3)	Rel. energy density
Diesel	14.5	1.38	3.79	0.99
Methane	17.2	1.24	4.30	0.89

The heating value of a substance, fuel in this case, is the amount of heat released during the combustion of a specified amount of it. The energy value is a characteristic of each substance. It is measured in units of energy per unit of the substance, usually mass. The most common unit is MJ/kg [6]. Lower and higher heating values of diesel and natural gas are showed in table 2.2.

In particular, the *lower heating value*, also called net calorific value, is the amount of heat released by a specified quantity of fuel, initially at 25°C, returning the temperature of the combustion products to 150°C, when it is assumed that latent heat of vaporization of water in the reaction is not recovered [7].

The *higher heating value*, also called gross calorific value, is the amount of heat released by a specified quantity of fuel, initially at 25°C, after combustion is done and the products return to the initial temperature (25°C). Here latent heat of vaporization of water in the combustion products is taken into account [7].

Table 2.2: Lower and heating values [7].

Fuel	Lower heating value (MJ/kg)	Higher heating value (MJ/kg)
Diesel	42.79	45.76
Natural Gas	47.14	52.23

The Wobbe index is also an interesting indicator. It shows the interchangeability of gaseous fuels. It relates to a comparative measure of thermal energy flow through a given size of orifice. Gases with similar Wobbe index can replace each other without varying the relative air-fuel ratio, working at the same metering settings. Wobbe index of methane is 48.17 MJ/m^3 [6].

2.1.4. Comparison with other alternatives and conclusion

Analyzing the different types of alternative fuels that are being developed and used at the moment, a comparison table between all of them has been realized in this thesis, in order to understand their advantages and disadvantages in a better manner. It must be taken into account that the comparison is made in respect of common fossil fuels, gasoline and diesel.

Biodiesel and Renewable diesel have been differenced. Biodiesel, also known as Fatty-acid methyl ester (FAME), refers to the “traditional” biofuel, which are usually manufactured from rapeseed oil, rape oil or soybean oil. Renewable diesel is hydrotreated vegetable oil (HVO), and it can be manufactured from any organic biomass, such as vegetable oil and a variety of waste and residues. Chemical composition of HVO is similar to fossil diesel, so it can be blended in all proportions achieving mixtures of high quality. However, when blending FAME biodiesel, it results in a mixture of a lower quality.

Table 2.3: Comparison table between different alternative fuels.

Fuel	Availability and cost	Storage and Transportation	Emissions	CO_2
Natural gas	High availability Low cost	Special deposits for storage (high pressure) Not very economical for long distance transport	Methane slip Low NO_x, CO, HC and SO_2	Low
Biodiesel	Quite low availability Expensive	Storage problems due to its corrosivity Limited infrastructures of distribution	Low CO, HC, SO_2 High NO_x	Low
Renewable diesel	Low availability Expensive	Compatible with existing diesel distribution infrastructure (pipelines and storage tanks)	Low CO, HC, SO_2 Slightly low NO_x	Low
Methanol	High availability Low cost	Stored easily but corrosivity problems Easy distribution	Low NO_x, CO, HC and SO_x	CO_2 from methanol obtainment
Ethanol (mainly E85)	High availability Expensive (low fuel economy)	Risk of flammability and corrosion implies special storage and carrying	Low CO and benzene (human carcinogen) High toxic acetaldehyde	Slightly low
Butanol	Low availability Expensive	It can be transported through the existing petroleum pipelines	Low CO and NO_x Higher HC	Low
Hydrogen	High availability Expensive	High energy obtainment process and difficult storage of big amounts Expensive and low efficient transportation	Zero emission but low NO_x	None
Propane (LPG)	Quite low availability Expensive	Huge network of transport, storage and distribution	Methane from its obtainment	Slightly low
Electricity	Low availability Expensive vehicles and batteries	Easy transportation, but battery storage technology needs further development	No emissions Gases produced during electricity obtainment	Slightly CO_2 during electricity obtainment

Taking a look at the table 2.3, it can be concluded that natural gas is an advantageous fuel alternative to study. Regarding to storage and transportation, it is noticeable that most of these alternative fuels present storage problems due to the corrosivity, as well as limited infrastructures of distribution. But it is mainly because they are currently being developed, so diesel and gasoline have a large advantage over them.

Its availability and low cost, as well as the reduction in terms of emissions, especially greenhouse gases such as CO_2, provide natural gas the interest of being researched in this master´s thesis as the main fuel in dual-fuel mode.

2.2. Dual-fuel

2.2.1. Dual-fuel history

Despite dual-fuel concept is not very well-known, first experiments concerning dual-fuel combustion took place in 1929 by Cave, and in 1930 by Helmore and Sokes. They tried to use hydrogen as a secondary fuel in diesel engines, saving the approximate quantity of 20% diesel. Even though it was a really effective discovery, in those years dual-fuel engines were not able to be commercialized due to its mechanical complexity and rough running caused by knocking and autoignition at relatively low compression ratios [8].

The first commercial dual-fuel engine was produced in Great Britain in 1939 by National Gas and Oil Engine Co., by using town gas or other type of gaseous fuels for the combustion. They worked with a really simple operation and were used in some regions where cheap stationary power production was required [8].

Later, over the Second World War, some diesel engines were modified in order to get dual-fuel engine vehicles, using in that case different gaseous fuels, such as coal gas, methane or sewage gas. Subsequent to these first researches made over the World War, dual-fuel engines have been developed and employed in several stationary power production applications, as well as road and marine transport (trucks, buses or ships) [8].

Currently, applications of dual-fuel systems can be found in engines used for the automotive sector, marine enforcements, electric generation, stationary engines, pumps and compressors. It is especially used in electric generation when control systems work in a particular way, trying to keep the rotation speed constant. Its use in automotive is still being improved at the moment, so only few dual-fuel engines can be found on the on-road market now [9].

Dual-fuel technology has caught attention of researchers because it is ecological and efficient in view of crude oil depletion and environmental pollution.

2.2.2. Dual-fuel concept

When talking on reciprocating internal combustion engines, it is known they can be divided into two different main groups attending to the combustion process type:

- Spark-ignition engines (SI).
- Compression-ignition engines (CI).

Spark-ignition engines, also known as Otto engines, are characterized by the external energy contribution to the thermodynamic cycle through a spark, in order to get the combustion start. A turbulent flame develops from the spark discharge, propagates across the premixed air – fuel mixture and extinguishes at the combustion chamber wall. In these engines the air-fuel mixture can be done outside the cylinder along the intake process. Another option is that the air is evidently introduced during the intake process, but fuel slightly later directly into the cylinder, when the compression starts, forming a heterogeneous mixture [10].

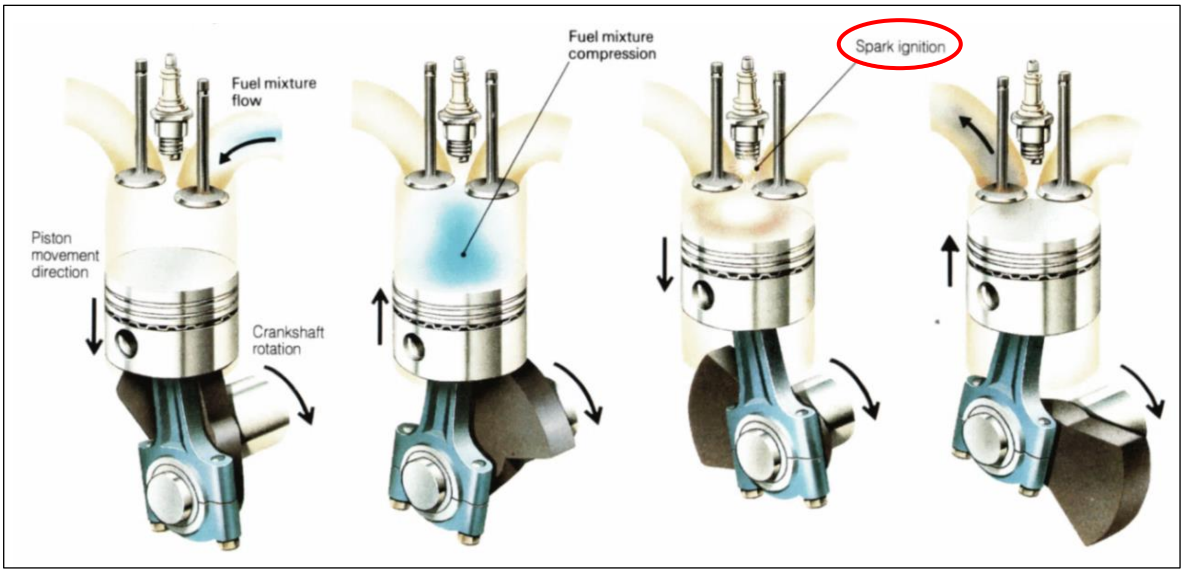

Figure 2.1: Spark-ignition combustion.

Compression-ignition engines start the combustion through an autoignition process of the fuel-air mixture, reaching high temperatures in the combustion chamber due to the compression process. Air alone is introduced along the intake process and fuel is injected directly into the cylinder just before combustion is required to start, during the end of the compression and before top dead center, when the air gets high levels of temperature in order to cause the autoignition. The flame extends rapidly through the already mixed portion of fuel that has enough air to burn. Mixing between fuel, air and residual gases in the cylinder continues as the expansion process proceeds, accompanied by further combustion [10].

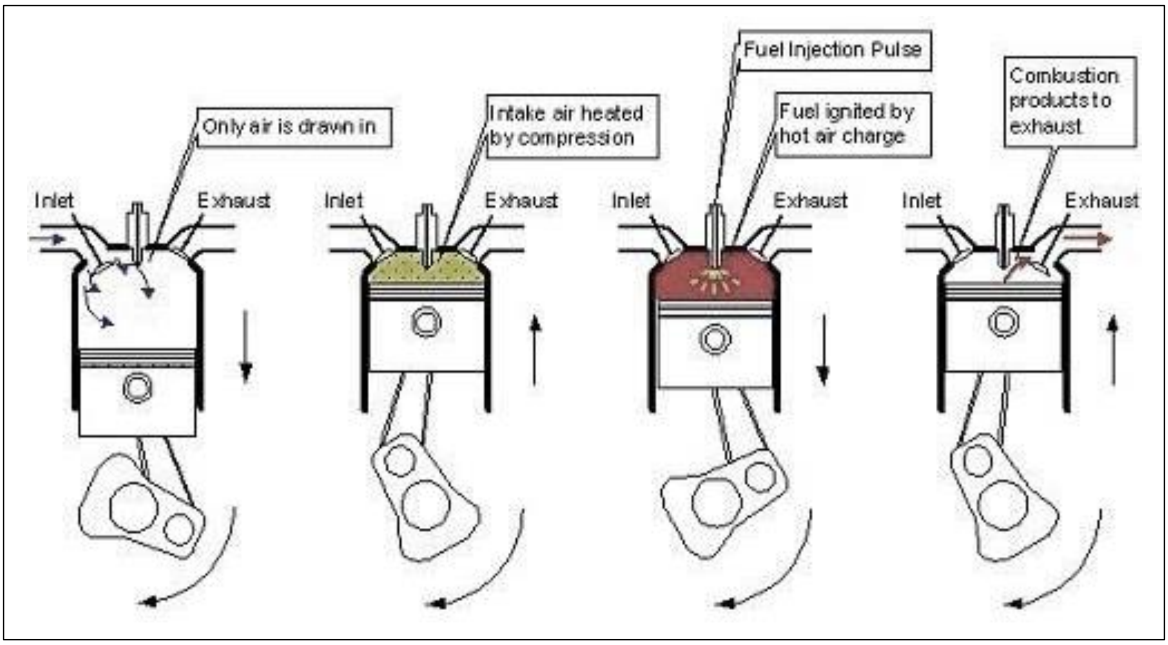

Figure 2.2: Compression-ignition combustion.

In dual-fuel engines, both types of combustion coexist together, considering diesel ignition as a “spark”. Natural gas is used as primary fuel in a compression-ignition type engine, but its autoignition temperature is extremely high and it is not reachable by its own. For this reason, a small amount of diesel injection serves as an ignition source, inducing the combustion and raising the temperature in the combustion chamber. The advantage of this engine is that it uses the difference of reactivity of the two used fuels. Besides, in case of lack of natural gas or other gaseous fuel, this type of engine is also able to run according to the pure diesel cycle [8].

Basic design of the diesel engine is not modified for dual-fuel use, as well as the principle of diesel combustion, compression ignition. Only changes that should be done are the addition of a gas injection system and an additional externally-fitted ECU. Temperatures and pressures in the cylinder remain in the limits of pure diesel operation. Nevertheless, diesel injectors in a dual-fuel engine ignite the compressed gas – air mixture in the cylinder, and electronically controlled gas injectors should be installed in the modified intake air manifold [11].

Obviously, combustion in a diesel engine is modified by the presence of natural gas, because it displaces intake air and changes the properties of the intake charge. For this reason, usually lower charge temperatures appear at the start of injection and an increase of the diesel ignition delay is shown, which results in NOx emission increment, but effect of natural gas substitution on emission depends on the engine type, operating mode, etc. [12].

2.2.3. Types of dual-fuel strategies

Despite several strategies of dual-fuel mode in compression ignition engines have been investigated, according to Shah et al. [12], there are two that are maybe the most important, in regard to the amount of diesel fuel that is replaced by the gaseous fuel: Conventional dual-fuel and Gas-Diesel direct injection system. Nevertheless, Conventional dual-fuel and Gas-Diesel are sometimes considered different concepts.

Conventional dual-fuel

In this case the diesel percentage can vary between 0.5% and 100%, so it corresponds with the type of higher amount of diesel [3,12].

Natural gas is introduced in the intake manifold mixing in that moment with the incoming air and being pushed into the combustion chamber. Then, when the cylinder goes up and the compression stroke is approaching to the top dead center, diesel fuel spray is injected directly into the chamber causing the burning due to the lower autoignition temperature of diesel. This type of strategy is recommended to be started with 100% diesel as a security measure. Knocking could happen specially when working with full load conditions if this is not done [12]. Its operation is shown in figure 2.3.

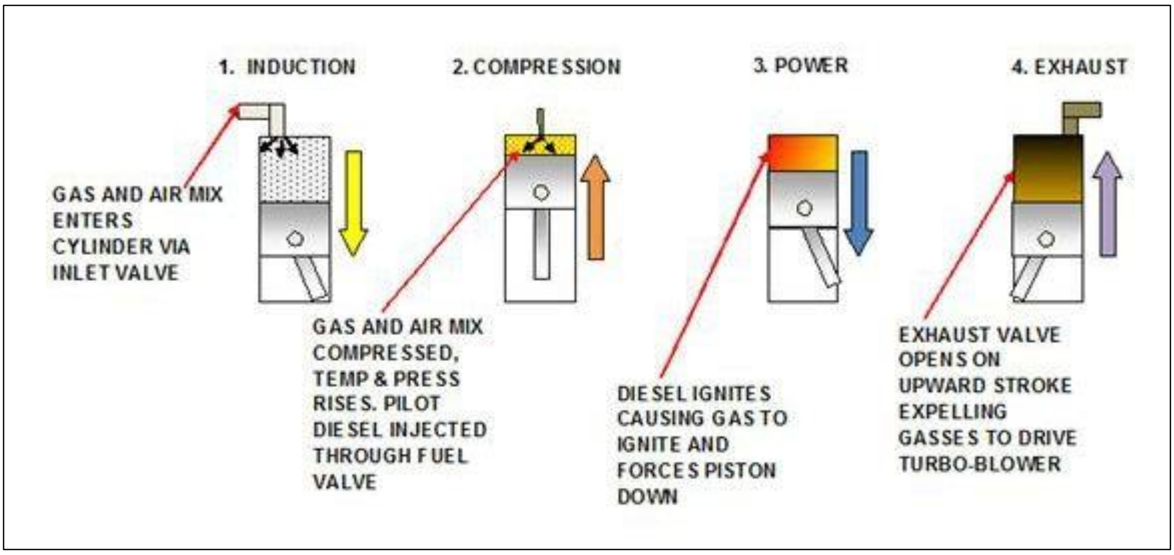

Figure 2.3: Conventional dual-fuel strategy.

In this dual-fuel mode it is important to be careful with the ignition delay of diesel fuel. The gaseous fuel introduction into the cylinder together with the intake air can cause variations in the mixture physical properties, e.g., lambda. Therefore, it could cause important changes in the temperature and pressure level at the top dead center, which can provoke some bad changes on the preignition reaction activity and energy release. Similarly, modifications in the gaseous fuel with the same amount of pilot diesel fuel and the same equivalent ratio can also produce serious changes in ignition delay. For example, it has been proved that propane, a more reactive gaseous fuel than methane, originates higher ignition delays than methane on dual-fuel combustion [12].

The advantage of this Conventional dual-fuel strategy is the benefit in terms of the Break Specific Fuel Consumption (BSFC), which measures the fuel efficiency as the rate of fuel consumption divided by the power produced, and also that it maintains the torque characteristics of the base diesel engine, sometimes with the necessity of increase bores, and reduces NOx and PM emissions in comparison with the diesel engine [3,12].

Gas – Diesel direct injection system

The amount of diesel in this second strategy is normally limited to 20%, so it is supposed to use smaller quantities than the conventional one [12].

The gaseous fuel works as the primary fuel and the pilot diesel injection plays the role of the ignition source as well, but the gas is injected directly in the cylinder with high pressure after the charged air has been compressed, like in typical diesel engines, which is also known as late cycle gas injection strategy. Evidently, a small amount of diesel injection is also required to ignite the natural gas – air mixture due to its poor autoignition quality, although the gaseous fuel injection is direct [12].

This strategy is not able to reduce NOx emissions as well as Conventional dual-fuel mode, power drops more and the running costs are higher [12]. Also PM are higher. However, it has the benefit that no methane slip remains in the cylinder, which leads to THC emissions decrease.

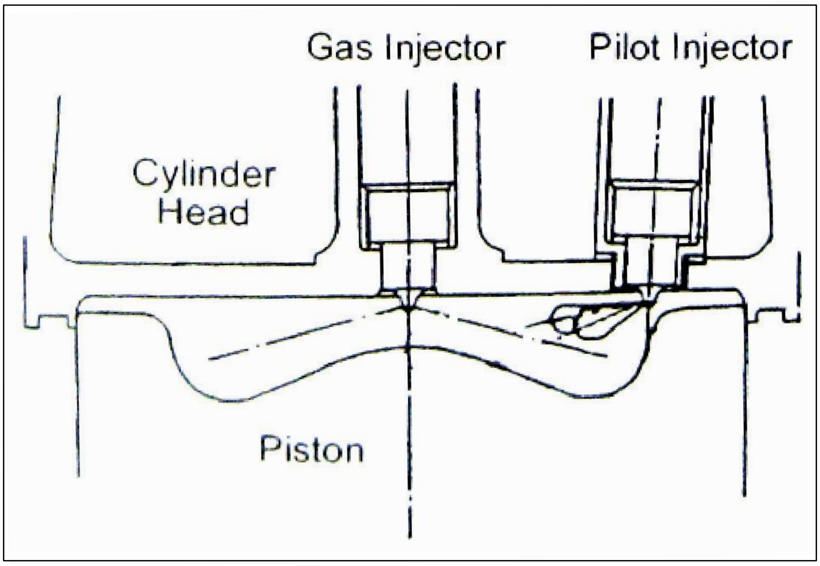

Figure 2.4: Pilot Ignited Natural Gas – Diesel strategy [12].

The distribution of the injector is different in this case, as it can be checked from figure 2.4. The diesel pilot injector is not situated at the center of the combustion chamber, but it is mounted as a side pilot injector, providing this mentioned small amount of diesel as an ignition source. In this case is the gas injector which is displaced in the center of the chamber, as it is the main fuel source [12].

Natural gas pressure injection is around 275 bar and it is injected just after some diesel pilot fuel, not allowing the injected gas to have time to be premixed with the surrounded air, so it is burned in the diffusion combustion mode [12].

This type of construction does not allow us to run the engine on pure diesel fuel operation, because of the emplacement of the side-mounted diesel injector. In order to overcome this problem, Bertrand D. Hsu developed the "H-Process", with a similar distribution but swapping the diesel pilot and the gas injectors [12]. It can be seen in figure 2.5.

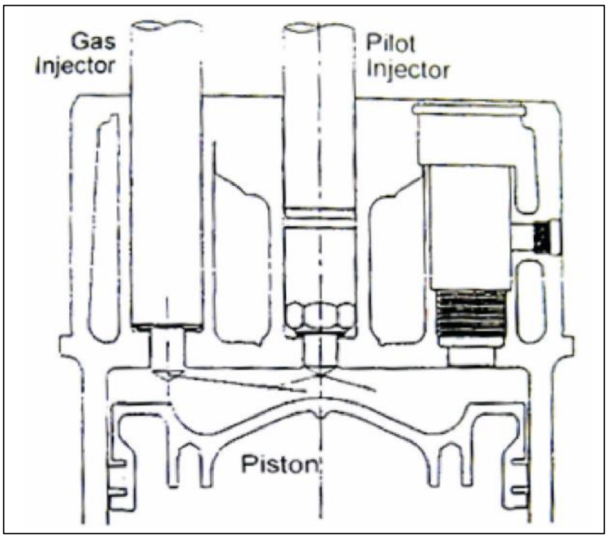

Figure 2.5: "H-Process" injectors position [12].

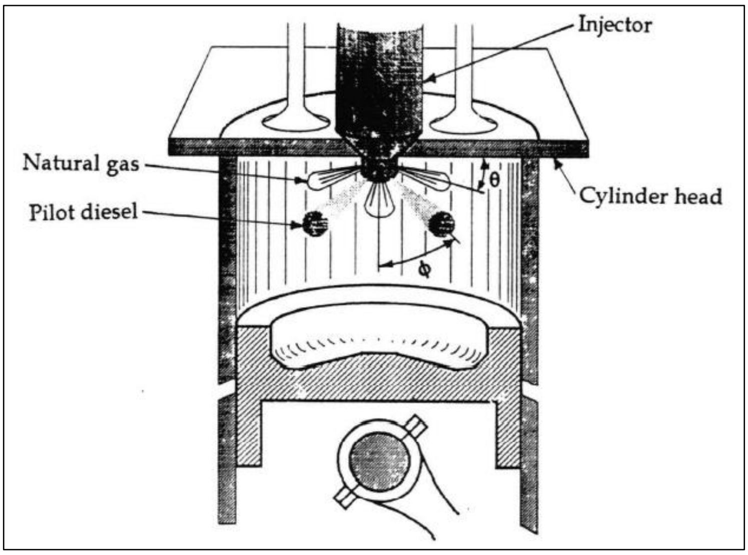

Figure 2.6: Combined injector in Pilot Ignited Natural Gas – Diesel.

Another distribution for this mode could be seen in figure 2.6, where both injectors are located in a combined unit injector but the pilot diesel spray and natural gas jet are separated in the circumferential direction by the angle ϕ [12].

2.2.4. Dual-fuel combustion

In a dual-fuel engine most of the energy release comes from the combustion of the gaseous fuel, while only a small amount of diesel liquid fuel provides ignition.

Karim et al. [13] described the dual-fuel combustion process as proceeding in three stages after ignition in a CI engine. First stage is due to the combustion of approximately half of the pilot fuel and a small amount of gaseous fuel. The second is due to diffusive combustion of the remaining pilot diesel and the rapid burning of the gas in the immediate surroundings. And finally, the third one is due to flame propagation through the remainder of the cylinder charge.

Main determining factors of the combustion process in a dual-fuel engine are the spray and ignition characteristics of the diesel pilot injection, as well as the type of gaseous fuel used and its overall concentration in the cylinder charge [14].

Over the combustion, premixed methane-air mixture becomes to higher temperatures and pressures as the top dead center position is approached. Some reactions can proceed during the combustion process even in the absence of pilot injection. Karim et al. experiments [13] show, through computational modeling with detailed chemical kinetics for the oxidation reactions of methane in air, that the mean temperature of the charge can increase significantly as the top dead center position is approached due to the increase of chemical activity, being especially noticed with mixtures that are not very lean [14].

Reactions occurred, even without the pilot, may produce partial oxidation products such as radicals, aldehydes and carbon monoxide, which can appear in higher amounts in the last part of the compression stroke, influencing the ignition and the combustion process of the pilot. These will contribute to the combustion of some gaseous charge of the pilot and its surroundings, as well as to the rest of the charge indirectly. Turbulent flame propagation from the pilot ignition regions will not advance through the charge until the concentration of gas is over a limiting value, function of the fuel employed and the operating conditions [14,15].

The combustion energy release of a dual-fuel engine can be studied as a combination of three overlapping components, as said some paragraphs before for the combustion process. In figure 2.7, the first stage (I) shows the combustion of the pilot. The second (II), represents the combustion of the gaseous fuel portion which is in the environs of the ignition and combustion centres of the pilot. And finally, the third (III), appears because of any preignition reaction activity and succeeding turbulent flame front [14].

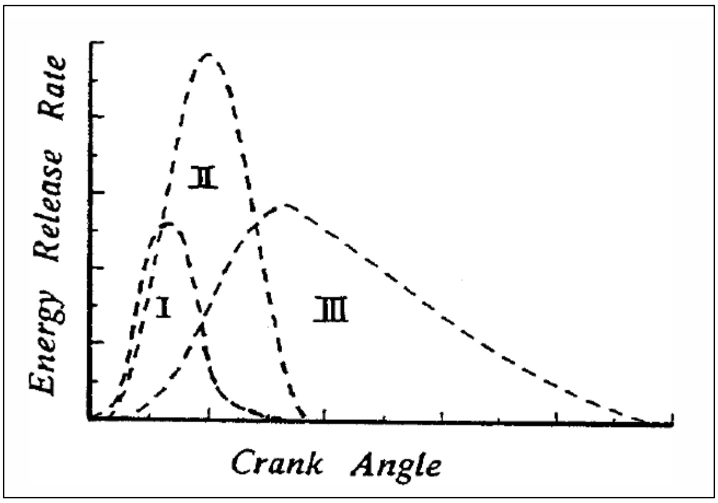

Figure 2.7: Components of the energy release rate in a dual-fuel engine [14].

2.2.5. Dual-fuel benefits

Dual-fuel engines present some benefits in comparison with the standard diesel and spark-ignited natural gas ones, especially in terms of power, efficiency, emissions and cost savings.

Regarding to performance and efficiency, dual-fuel technology closely matches diesel power, torque and efficiency. In addition, it beats standard diesel power with fewer air quality pollutants and less greenhouse gas emissions in terms of CO_2 [11]. On the other hand, of course, dual-fuel performance implies higher amounts of CH_4 in the form of methane slip, which is, as known, another important contributor of the greenhouse effect.

Figure 2.9, provided by Clean Air Power [11], illustrates the transition of CO_2 saving over five years, based on a truck running 156,000 miles a year, with 60% gas substitution rate.

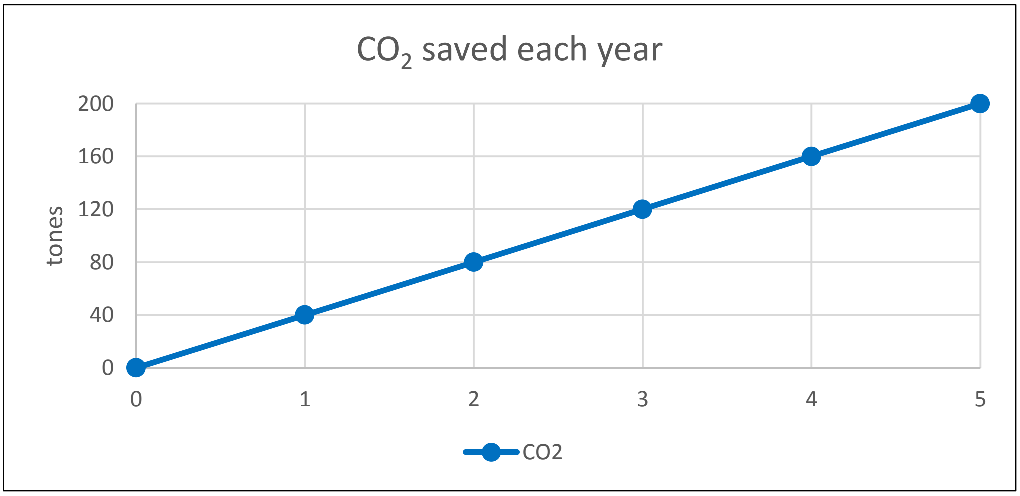

Figure 2.8: CO_2 saving evolution during 5 years [11].

Regarding to cost savings, it is remarkable that dual-fuel engines have better fuel economy than a dedicated natural gas engine, and the average in-service gas substitution rates are within 60 and 80 %, which means less diesel quantities and less money cost [11].

A cost saving evolution from a UK truck, running 156,000 miles a year with a 60% gas substitution rate, can be seen in figure 2.10.

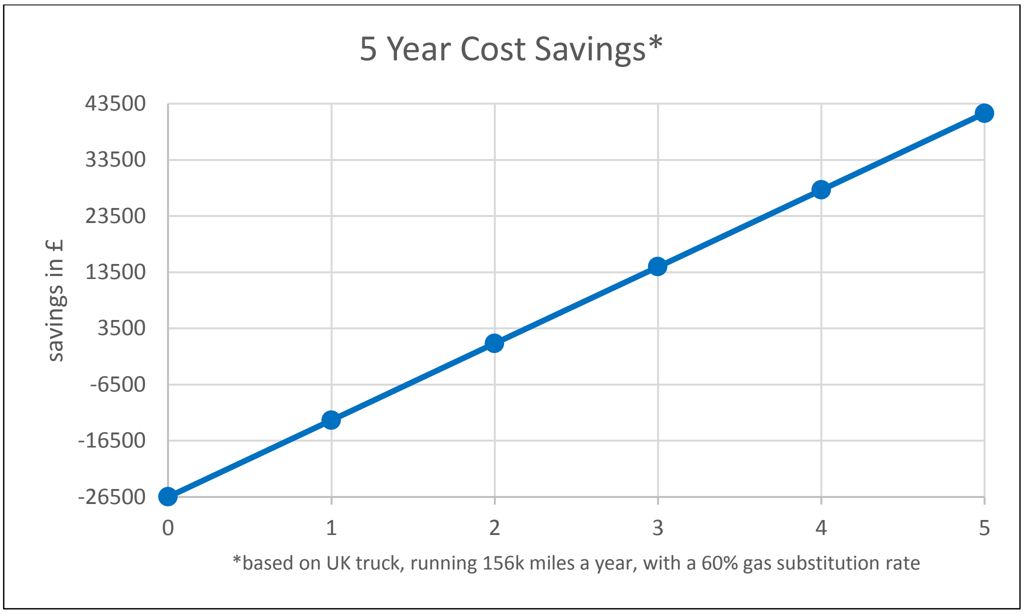

Figure 2.9: Cost saving evolution during 5 years [11].

It can be concluded that Dual-fuel engines take the benefits of the efficiency of compression ignition engines while dramatically cutting consumption of diesel fuel, which can mean an important saving of money if sufficient substitution rate is got. In addition, it gives to us a valuable fuel flexibility, as a dual-fuel engine can run on 100 percent diesel if gas is not available in the moment, as well as it can operate on liquefied natural gas (LNG) or compressed natural gas (CNG) [11].

2.3. Emission formation

Probably the main concern for developers in the field of internal combustion engines is focused in the emissions. It is driven by the emission legislation and all the efforts done in researches are directed to achieve the established limits. Developing an engine consists on finding the optimum between engine fuel consumption, development costs and production costs, meeting at the same time the legislated emission values.

Emissions from both sparked-ignited and compression-ignited engines can be classified into two different categories: global and local emissions. The differences lie in the scope they have according to the environmental concern [16].

Global emissions

These emissions are the ones that affect the whole planet. Existing legislation and standards for engines do not generally take into account global emissions, except some recent CH_4 limitations, because most of the attention of researchers goes for local emissions.

This type is mostly formed by CO_2 emissions in any kind of engine, and it is not regulated because it is usually related to a poor fuel economy. That is the reason why developers try to minimize them as much as possible without the necessity of being forced to it, just to get a competitive product in the market [16].

Carbon dioxide (CO_2) and methane (CH_4) are the most harmful global emissions produced during dual-fuel combustion.

The formula for a general combustion reaction is:

$$C_xH_y + \left(x + \frac{y}{4}\right) O_2 \rightarrow xCO_2 + \left(\frac{y}{2}\right) H_2O + energy \tag{2.1}$$

As seen, carbon dioxide, CO_2, is an inevitable product of the hydrocarbon combustion. It is an inert gas so it does not affect to plants or animals. The problem is that it is a greenhouse gas, so it is one of the gases that contribute to keep the Earth within adequate temperatures. High amounts of this gas contribute to the phenomenon known as greenhouse effect, which consists in the reduction of the heat emission to the space and, consequently, a higher planet warming. This temperature rising is called “global warming”.

On the other hand, methane should be considered. The presence of CH_4 in the exhaust gases is called methane slip, that is, unburned methane. In terms of global warming, it is 21 times worse than CO_2, making methane a very potent greenhouse gas. It means if it is not carefully treated, methane slip would beat dual-fuel engines compared to gasoline or diesel ones.

Local emissions

In this case, the emissions only affect negatively to the closest environment to the source. Nitrogen oxides (NOx), unburned hydrocarbons (HC), carbon monoxide (CO) and soot emissions are the most common emissions of this group.

2.3.1. Nitrogen oxides

Nitrogen oxides (NO_x) are mostly formed by nitric oxide (NO) and nitrogen dioxide (NO_2). NO is the most common in engine emissions, although some NO can be easily converted into NO_2 in certain conditions of low temperature and large amounts of oxygen [16,17,18].

The scope of nitrogen oxides formation in internal combustion engines basically depends on the local conditions in the cylinder, especially on the concentrations of oxygen and nitrogen, and being extremely sensitive to local temperatures [17].

The most propitious conditions for NO_x formation during the combustion are achieved when the local temperature rise and, at the same time, an adequate amount of oxygen and nitrogen is available. In dual-fuel engines the availability of oxygen and nitrogen is guaranteed, since this type of engines usually works under lean mixture conditions, which means that NO_x formation is mainly governed by local temperature [17].

The harmfulness of nitrogen oxides is due to its ability to form, in the presence of other compounds, the tropospheric ozone and the photochemical smog, as well as the acid rain [18].

Tropospheric ozone, also called environmental ozone, is a greenhouse gas created from photochemical reactions between nitrogen oxides (NOx), carbon monoxide (CO) and volatile organic compounds (VOCs), derived from combustion processes. It constitutes a problem since ozone, in sufficient concentration, can provoke damages to human health as well as plants. At the same time, it contributes to the Earth surface warming.

Photochemical smog is also formed by reactions between NO_x and VOCs (primary pollutants), accompanied of some secondary pollutants. It is produced when the primary pollutants react in the atmosphere with sunlight to form the secondary ones, which also combined with the primary emissions again form the mentioned smog.

Acid rain appears when sulfuric dioxide or nitrogen oxide reacts with water molecules in the atmosphere, producing acids that, in high levels, can be harmful for plants, animals and infrastructure.

There are three main mechanisms of NO_x formation: *Thermal* NO_x*, Prompt or Fenimore* NO_x *and Intermediate* N_2O *formation mechanism*.

Thermal NO_x is the most important mechanism at high temperatures. It is due to the atmospheric N_2 oxidation, as nitrogen is the most common oxidant on combustion. NO_x formation from N_2 happens through a chain reaction mechanism called extended Zeldovich mechanism. Similarly to diesel engines, the main reactions that guide the NO formation in a dual-fuel engine are:

$$N_2 + O \leftrightarrow NO + N \tag{2.2}$$

$$N + O_2 \leftrightarrow NO + O \tag{2.3}$$

$$N + OH \leftrightarrow NO + H \tag{2.4}$$

The name thermal NO_x refers to the high temperature dependence of this mechanism, due to the high activation energy of the first, rate-limiting reaction. It is believed that thermal NO_x is responsible of around 90% of the NO_x emissions [16,18,19].

Prompt or Fenimore NOx mechanism is based on the reaction of certain hydrocarbon compounds from fuel decomposition with N_2, forming amines or cyano compounds, which rapidly convert into intermediate substances that end in NO formation. Schematically, the mechanism can be described by the following reactions:

$$CH + N_2 \leftrightarrow HCN + N \tag{2.5}$$

$$C + N_2 \leftrightarrow CN + N \tag{2.6}$$

$$CH_2 + N_2 \leftrightarrow HCN + NH \tag{2.7}$$

$$CH_2 + N_2 \leftrightarrow H_2 + CN + N \tag{2.8}$$

Reaction (2.5) is the dominant and NO is obtained from the reaction of nitrogen atoms produced in the previous reactions, with OH radicals, following (2.4). Contribution of this mechanism to the total NO emission is estimated below 5% [3,21].

Intermediate N_2O formation mechanism is important during NO formation processes at low temperatures and lean mixtures. One of the most known mechanisms for this case is the one developed by Lavoie et al. [15]:

$$N_2O + O \leftrightarrow NO + NO \tag{2.9}$$

$$N_2O + O \leftrightarrow N_2 + O_2 \tag{2.10}$$

$$N_2O + H \leftrightarrow N_2 + OH \tag{2.11}$$

This N_2O consideration in the combustion process is especially important in CI engines employing new strategies of NO_x reductions, such as EGR, since combustion temperature decrease provoked by these strategies limits the NO formation but increments N_2O, being this species an important collaborator of the greenhouse effect [19].

2.3.2. Unburned hydrocarbons

Hydrocarbons emitted into the atmosphere are suspected to be highly carcinogenic, especially the polycyclic aromatic hydrocarbons. They can cause photochemical smog and be prejudicial for human beings and animals. HC emissions from a dual-fuel operation combine SI and CI engines contributions [19].

HC from CI engines are basically derived from the lack of oxygen or the existence of cool walls which cause an incomplete combustion, as well as some diesel fuel remaining in the injector sac, but these contributions are insignificant in comparison with SI engines [16,20].

In SI engines, where there is a premix prior to the ignition, the three main sources of HC emissions are: *crevice losses, adsorption and quenching* [16].

Crevice losses perform the largest contribution. Portions of the premixed charge are driven into narrow and cold regions, in which conditions are unfavorable for flame propagation. Then, when expansion occurs and the cylinder pressure drops, the unburned compounds from the crevices return to the combustion chamber and are scavenged together with the exhaust gases [16].

Adsorption phenomenon is similar. Hydrocarbons, at high pressure, are adsorbed into the oil film of the cylinder walls and the deposits in the combustion chamber. When expansion and cylinder pressure drop occur, the adsorption reverses and these HC are back into the combustion chamber itself [16].

Quenching, on the other hand, is based on lower temperatures phenomenon. A boundary layer exists close to the cylinder wall, with lower temperature in comparison with the rest of the charge, causing the flame quenches before it reaches the wall. It makes some HC in the boundary layer to escape combustion [16].

2.3.3. Carbon monoxide

Carbon monoxide is toxic to humans and it can cause different symptoms, from light headaches to death, depending on the emitted amounts [16].

CO formation mechanism is an essential intermediate step on a hydrocarbon oxidation, appearing mainly with excessive rich mixtures, as well as lean mixtures at sufficient heterogeneous conditions [19].

In SI engines, CO is basically formed by the dissociation of the CO_2 molecules at high temperatures. However, at lean conditions, CO emissions can be due to low temperatures, similarly to CI engines [16,19].

In engines with diffusion combustion, CO appears in excessively lean regions of the combustion chamber in which combustion is slower, preventing the flame propagation and partially oxidizing carbon from the combustion. On the other hand, it can also be formed due to excessively rich regions, result of the deficient blended between the evaporated hydrocarbon and the air in the chamber, not giving time to oxidation reactions to be completed. This second happening usually occurs at full loads and lower excess air levels [19].

2.4. Emission legislation

Emission standards are legal required values established by governments in order to control the air pollutants released into the atmosphere. They are made with the objective of getting adequate air quality standards and protecting the human health.

These permissible amounts of specific air pollutants are studied differently depending on the sources which they come from, such as motor cars, heavy-duty vehicles or ships. Other non-vehicle emissions are also regulated, including for example industry, power plants or some other sources of air pollution.

Another classification is made based on the countries. Different standards can be found in each area of the world. This master´s thesis will be focused in the European Union, also mentioning the importance of the United States legislation.

The European Union has made its own emission legislation, Euro standards, and it should be met by all the new vehicles sold. Standards are set for on-road vehicles, trains, barges and off-road machines, but not for others like ships or planes. The stages are referred as Euro 1, Euro 2, Euro, etc., except for heavy-duty vehicles, which use Roman numbers instead of the Arabic ones, i.e., Euro I, Euro II, etc. They started to be applied from 1993 [21]. In off-road vehicle emission standards, the word “Euro” is replaced by “Stage”.

The US legislation, Tier standard, is made by the Environmental Protection Agency (EPA). National Tier 1 started to have effect in 1994, but there are also different standards according to different vehicles. California has his own vehicle emissions standards, set by the California Air Resources Board (CARB), and other states can choose between the national and California’s one. Due to the California’s automotive market is one of the biggest in the world, several states from the United States have chosen the CARB standards over the last few years. CARB has also great influence in EU emissions standards [21].

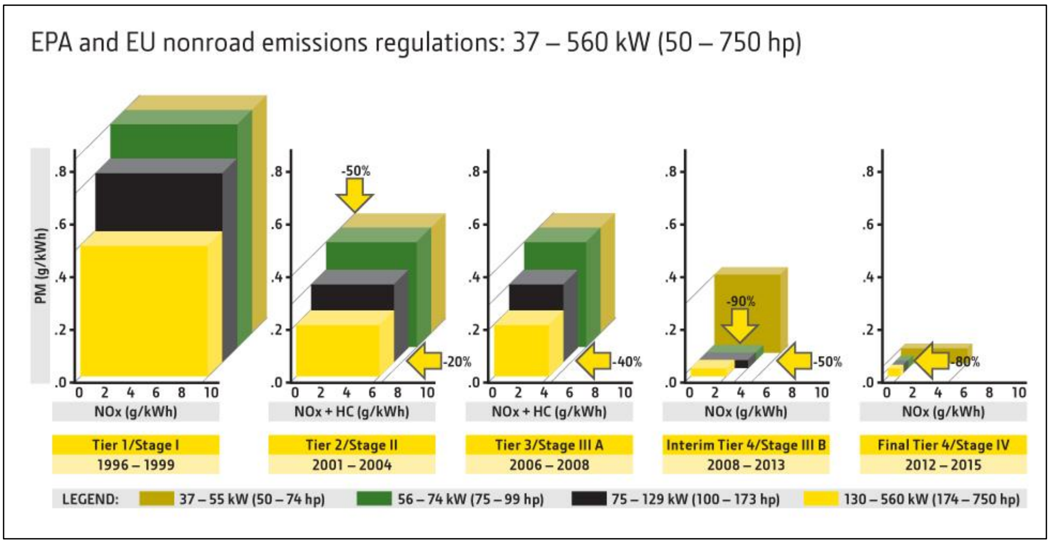

Figure 2.10: Evolution of EU / US emission standards for different power ranges.

At the present time, the emissions that are regulated by the European Union are: nitrogen oxides (NO_x), total hydrocarbons (THC), non-methane hydrocarbons (NMGC), carbon monoxide (CO) and particulate matters (PM). NO_x is one of the main worries on emissions, so figure 2.10 illustrates the evolution of the emission standards from Tier 1 and Stage I to Tier 4 and Stage IV, for non-road heavy-duty compression-ignition vehicles, which are the closest to the engine used for the realization of this master´s thesis [21].

The European Union imposes emission limits for lots of applications of internal combustion engines. There are different standards depending on the type of engine and vehicle, many of which have some new legislative steps being prepared.

The list of different kinds of vehicles or engines includes:

- Passenger cars and light commercial vehicles
- Heavy-duty engines for trucks and buses
- Motorcycles, tricycles, quadricycles and mopeds
- Engines for non-road mobile machinery
- Engines for agricultural and forestry tractors
- Propulsion engines for recreational craft and personal watercraft
- Combustion heaters for motor vehicles and their trailers

Apart from mobile sources, there are also some emission limits covering:

- Large combustion plants
- Waste incinerators
- Vapor emissions from fuel distribution and storage
- Other manufacturing operations

2.4.1. Evolution of the Euro legislation

In order to get a view of the evolution of the European emission legislation over the past twenty years, the best manner to do it is focusing on the passenger cars legislation, because it is the most common and known one. There are also some differences between diesel and gasoline vehicles, so this thesis will focus on diesel since it is closer to the research area studied.

Over the last 20 years there have been several researches in the emissions field and manufacturers have invested a lot in products which could reduce drastically the amount of pollutants coming out of the tailpipe into the atmosphere.

For an overview, in table 2.4 can be seen the tendency of NO_x and PM emission standards reduction during the different stages of the EU legislation.

Table 2.4: EU emission standards Euro 1 – Euro 6 measured in g/kWh [22].

Euro standard	Date enforced	NOx	PM
Euro 1	1993	8.0	0.36
Euro 2	1996	7.0	0.15
Euro 3	2001	5.0	0.10
Euro 4	2006	3.5	0.02
Euro 5	2009	2.0	0.02
Euro 6	2014	0.46	0.01

The changes from Euro 5 to Euro 6 alone are more-than-impressive with the new legislation demanding a drop in NO_x of 75% compared with the previous standard, while changes to the procedure that particulate matter is counted will see a drop in PM of over 90% [22].

Finally, it is interesting to foreground the evolution differences between Off-road standards and On-road or Or-highway standards. The stages in the second one are more developed, which is reasonable since most of the engines are used in on-road vehicles.

Diesel Progress International published a diagram (figure 2.11) in which can be seen the different evolution in both cases.

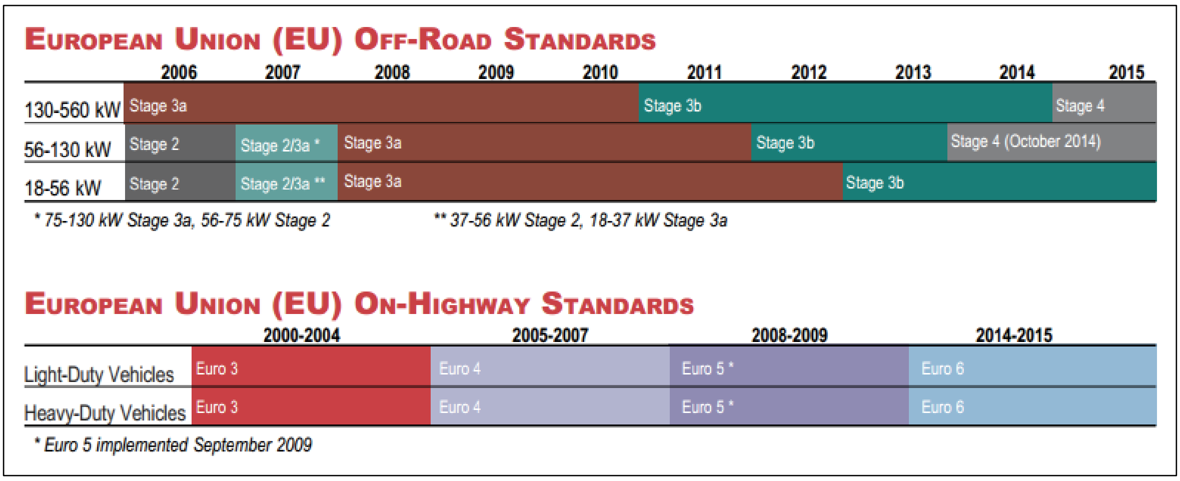

Figure 2.11: EU off-road and on-road standards 2000-2015 [23].

2.4.2. Emission standards of non-road heavy duty C.I. engines

As it has been introduced in previous figure 2.11, regarding to the off-road standards, the last developed stage is Euro IV. In this section can be seen the exact emission value limits in terms of specific emissions (g/kWh) for the most important pollutants, including CO, HC, NO_x and PM, during all the stages of the EU legislation.

Thanks to the Worldwide Emissions Standards specified by Directive 97/68/EC and five amending Directives adopted from 2002 to 2012, non-road mobile machinery emission limits between stages I and IV can be seen in tables 2.5, 2.6 and 2.7. These values are included in ECE GTR No. 11, a test procedure for compression-ignition engines to be installed in agricultural and forestry tractors and in non-road mobile machinery with regards to the emissions of pollutants by the engine [21].

GRT 11 is applicable to the determination of gaseous and particulate mass exhaust emissions from compression-ignition engines with power ranges between 19 and 560 kW to be used in non-road mobile machinery and tractors [21].

The first European off-road mobile equipment legislation was enacted on 16th of December in 1997. These initial regulations were included in two stages: Stage I implemented in 1999 and Stage II implemented between 2001 and 2004, with some differences according to the engine power output [21].

Table 2.5: Euro I and Euro II for non-road heavy duty C.I. engines [24].

Cat.	Net power	Date*	CO	HC	NO_x	PM
	kW		*g/kWh*			
Stage I						
A	130 ≤ P ≤ 560	1999.01	5.0	1.3	9.2	0.54
B	75 ≤ P ≤ 130	1999.01	5.0	1.3	9.2	0.70
C	37 ≤ P ≤ 75	1999.04	6.5	1.3	9.2	0.85
Stage II						
E	130 ≤ P ≤ 560	2002.01	3.5	1.0	6.0	0.2
F	75 ≤ P ≤ 130	2003.01	5.0	1.0	6.0	0.3
G	37 ≤ P ≤ 75	2004.01	5.0	1.3	7.0	0.4
D	18 ≤ P ≤ 37	2001.01	5.5	1.5	8.0	0.8
*Stage II also applies to constant speed engines effective 2007.01						

Stages III and IV were promulgated by the European Parliament on the 21st of April in 2004, although for agricultural and forestry tractors it happened on the 21st of February in 2005. Stage III was divided into Stages IIIA and IIIB, and it was used from 2006 to 2013, entering Stage IV in 2014 [21].

Between Euro II and Euro III an important reduction can be noticed especially in NO_x emissions, as well as HC, and, from Euro III B, also PM.

Table 2.6: Euro III A and Euro III B for non-road heavy duty C.I. engines [24].

Cat.	Net power	Date*	CO	HC	HC+NO_x	NO_x	PM
	kW		*g/kWh*				
Stage III A							
H	130 ≤ P ≤ 560	2006.01	3.5	-	4.0	-	0.2
I	75 ≤ P ≤ 130	2007.01	5.0	-	4.0	-	0.3
J	37 ≤ P ≤ 75	2008.01	5.0	-	4.7	-	0.4
K	19 ≤ P ≤ 37	2007.01	5.5	-	7.5	-	0.6
Stage III B							
L	130 ≤ P ≤ 560	2011.01	3.5	0.19	-	2.0	0.025
M	75 ≤ P ≤ 130	2012.01	5.0	0.19	-	3.3	0.025
N	56 ≤ P ≤ 75	2012.01	5.0	0.19	-	3.3	0.025
P	37 ≤ P ≤ 56	2013.01	5.5	-	4.7	-	0.025
*Dates for constant speed engines are: 2011.01 for categories H,I and K; 2012.01 for category J							

Table 2.7: Euro IV for non-road heavy duty C.I. engines [24].

Cat.	Net power	Date	CO	HC	NO_x	PM
	kW		*g/kWh*			
Q	130 ≤ P ≤ 560	2014.01	3.5	0.19	0.4	0.025
R	56 ≤ P ≤ 130	2014.10	5.0	0.19	0.4	0.025

On September 25, 2014, the Stage V was proposed by the European Commission, introducing some important changes, and specifying emission requirements for all categories of compression-ignition engines for non-road engines, replacing Directive 97/68/EC and its amendments [21].

Stage V includes some regulations for a large variety of machines, going from minute gardening and handheld equipment to large engines used in cars or locomotives. In comparison with the previous stages, the scope is more extended, covering also engines below 19 kW that were previously unregulated. It includes as well a limitation in the number of particle number (PN) emissions, being a complementation of the current particle mass (PM) limit, which allows ultrafine particles to be limited too. This stage will also be the first one that includes some in-use testing for non-road mobile machinery. The details of the in-use testing proposal will need to be met from December 31, 2016 [21].

The initial plan for the introduction dates of this new stage is:

- Initial date for engines under 19 kW: January 1, 2019.
- Initial date for all other C.I. engines except the ones in the range 130-560 kW: 3 years after.
- Initial date for engines between 130 and 560 kW: 4 years after.

In table 2.8, stage V emission standards for non-road engines are presented, covering the power range from 0 to 56 kW, as well as all types of engines above 56 kW.

Table 2.8: Euro V for non-road heavy duty C.I. engines [24].

Cat.	Net power	Date	CO	HC	NO_x	PM	PN
	kW		*g/kWh*				*1/kWh*
NRE-v/c-1	P < 8	2019	8.00	7.50[a,c]		0.40[b]	-
NRE-v/c-2	8 ≤ P < 19	2019	6.60	7.50[a,c]		0.40	-
NRE-v/c-3	19 ≤ P < 37	2019	5.00	4.70[a,c]		0.015	1×10^{12}
NRE-v/c-4	37 ≤ P < 56	2019	5.00	4.70[a,c]		0.015	1×10^{12}
NRE-v/c-5	56 ≤ P < 130	2020	5.00	0.19[c]	0.40	0.015	1×10^{12}
NRE-v/c-6	130 ≤ P < 560	2019	3.50	0.19[c]	0.40	0.015	1×10^{12}
NRE-v/c-7	P > 560	2019	3.50	0.19[d]	3.50	0.045	-

[a] HC + NO_x
[b] 0.60 for hand-startable, air-cooled direct injection engines
[c] A=1.10 for gas engines
[d] A=6.00 for gas engines

For engine categories where an A factor is defined, the HC limit for fully and partially gaseous fueled engines indicated in the table replaced by the one calculated from the formula: HC = 0.19 + (1.5 × A × *GER*); where *GER* is the average gas energy ratio over the appropriate cycle.

As said before, Stage V regulations would introduce a new limit for particle number emissions. The PN limit is designed to ensure that a highly efficient particle control technology, such as wall-flow particulate filters, be used on all affected engine categories. The Stage V regulation would also tighten the mass-based PM limit for several engine categories, from 0.025 g/kWh to 0.015 g/kWh [21].

2.5. Exhaust Gas Recirculation

2.5.1. EGR concept

As commented, the reduction of the environmental impact of combustion processes represents an arduous challenge in automotive technology. Particularly recirculation of exhaust gases has become crucial in the last few years, although it is not a new technique, in order to try to reduce the emission levels. This method has been crucial in the achievement of the worldwide standards.

Current environmental concern could have promoted that European directive proposals have established stricter limits of NOx emissions during the last years, especially because this reduction has become one of the most difficult goals to obtain. It is due to the effect caused by the rest of technologies used in this area, such as supercharging, which tend to provoke the opposite effect. In addition, new exhaust gas recirculation valves have been developed recently as well as better electronic controls, which allow a better EGR precision and shorter response periods.

Exhaust gas recirculation (EGR) has a clear mission: reduce pollutant NO_x emissions that leave to the atmosphere through the exhaust pipe. The recirculated burned gases work as a diluent in the unburned mixture. Absolut temperature reached after combustion goes inversely with the burned gas mass fraction. The combined effect of the highest specific heat than the air due to the CO_2 and H_20 concentration, and the reduction of the flame propagation speed caused by the presence of inerts, leads to a significant reduction of flame temperature and, thus, combustion temperature. It causes the reduction of NO_x levels [19,25]. However, it also reduces the combustion rate and, therefore, makes the achievement of a stable combustion more difficult [10].

This exhaust gas recirculation has a positive effect on the nitrogen oxides emissions due to this reduction of the combustion temperature, but, on the other hand, extremely high recirculation ratios could be negative in terms of engine durability [25]. It is mainly due to the growing formation of particulate matter when increasing EGR rate.

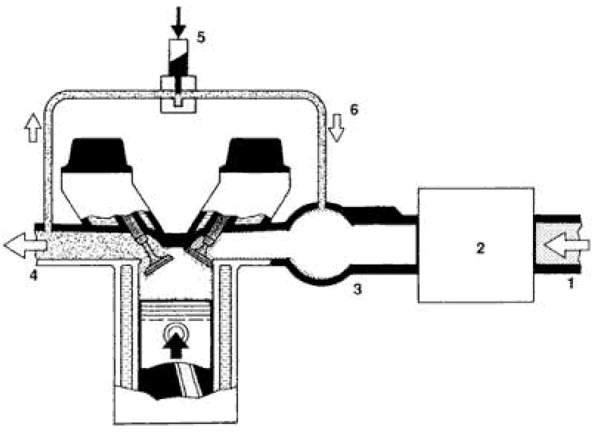

Figure 2.12: Schematic diagram of EGR [25].

Figure 2.12 shows a schematic diagram of the exhaust gas recirculation system, where the path followed by the exhaust gases can be appreciated since the combustion happens.

2.5.2. EGR classification

In order to understand the different methods that can be used to produce an exhaust gas recirculation, it is interesting to make a classification based on the characteristics that define the manner in which the EGR is introduced in the cylinder.

Depending on the place where the recirculated gases are introduced

- Internal EGR: Exhaust gases remains in the cylinder during the exhaust stroke through an adequate design of the cam profile and the proper valve timing [26].
- External EGR: Exhaust gases are introduced in the intake pipe by recirculation, being this method the most common between these two [26].

Depending on the pressure of the recirculation process

- Low pressure EGR: Exhaust gases are extracted after the turbine and introduced before the compressor (low pressure points of the circuit). In this case the circuit is basically at atmospheric pressure, which can cause some problems due to compressor deterioration [26]. A low pressure EGR system distribution can be seen in figure 2.13.
- High pressure EGR: Exhaust gases are extracted before the turbine and introduced before or after the intercooler, but typically after the compressor (high pressure points of the circuit). Occasionally, the different pressure across EGR line on this system can be a problem, so that some differential pressures need to be created artificially by altering the efficiency of a turbocharger or by introducing other auxiliary elements, such as venturi systems or some types of turbines with variable geometry [26]. This distribution is shown, as well, in figure 2.14.

Depending on the EGR temperature

- Cold EGR: Exhaust gases are cooled down so that the whole intake mass, air and recirculated gases, has a similar temperature as it would have without EGR. This type has the advantage of having a lower intake temperature, which means lower temperatures at the end of the compression stroke too, and this is beneficial in terms of NO_x emissions. The disadvantage is the requirement of a heat exchanger to cool down the gases [26].

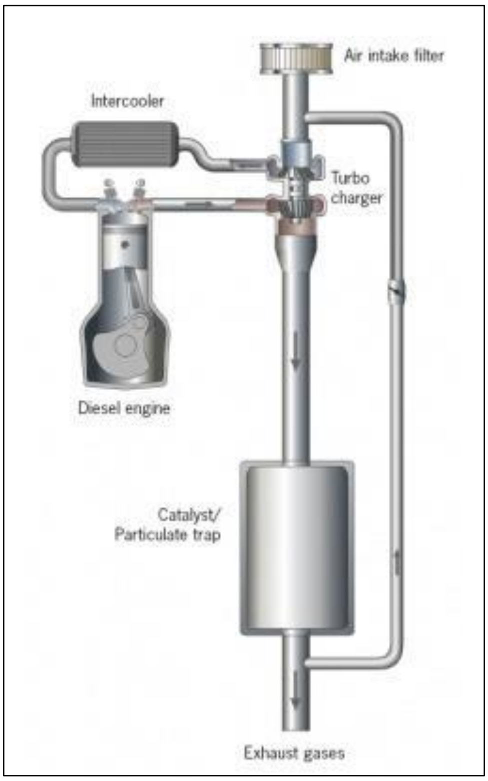

Figure 2.13: Low pressure EGR system.

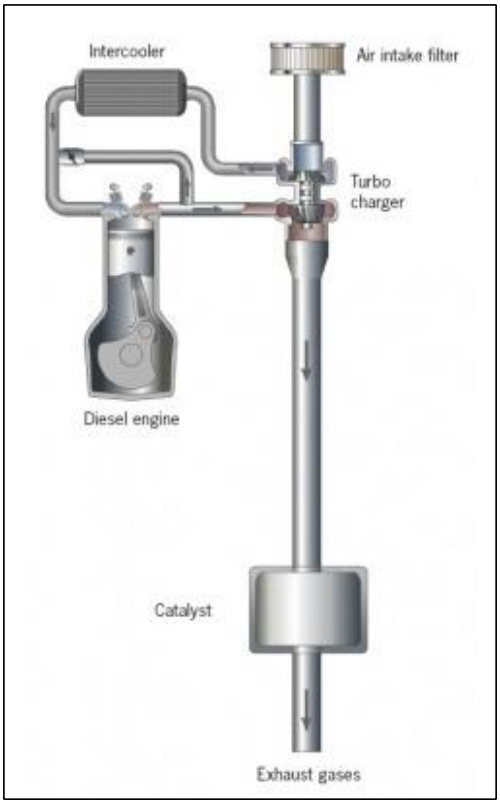

Figure 2.14: High pressure EGR system.

- Hot EGR: Exhaust gases are introduced in the intake with the same temperature as they had in the exhaust, so that the mass now is bigger than it would be without EGR. This one combines the NO_x-reducing effect with the improved flame propagation from higher inlet temperatures, allowing a possible simultaneous reduction of HC, CO and NO_x [16,26]. However, NO_x reduction with this method is more limited since temperatures reached during the combustion are higher.

Depending on the total mass admitted

- EGR of addition: The total mass of fresh air introduced in the cylinder keeps constant subsequent to adding EGR, so that the recirculated gases perform an additional mass. To use this method it is indispensable to increase the intake pressure, in the same proportion of the gases that want to be recirculated [26].

- EGR of substitution: The total mass admitted in the cylinder, air and recirculated gases, is constant, so that the EGR is replacing part of the fresh air mass that it would be without EGR [26].

In this study, a system based on an external, high pressure, cold and substitution EGR is used, with some of the theoretical background modified due to the fact that a research engine is utilized in this case.

2.5.3. EGR influence on combustion

As previously said, EGR is a popular technique of reducing NO_x levels in engines, and it is because of different phenomena produce during combustion as much in diesel and gasolines engines as in dual-fuel ones.

The exhaust gas acts as an inert gas in the combustion chamber, it does not participate in the combustion reaction. Nevertheless, this recirculated exhaust gas still contains a slight oxygen concentration, but this portion is very low compared with the oxygen amount in the fresh air. Therefore, the total oxygen concentration in the cylinder is reduced due to the replacement of fresh air by some recycled exhaust gas. The exhaust gases have a higher specific heat capacity than the fresh air, inasmuch as H_20 and CO_2 are contained. This performance results in a reduced combustion temperature and, thus, lower NO_x emissions, being an additional cooling of the recirculation helpful in regard to this effect [20,27,28].

It is suggested that disadvantages of using EGR in both SI and CI engines, such as noise levels or wear of piston rings due to relatively higher SO_2 concentrations, can be avoided in dual-fuel engines [28].

Likewise, dual-fuel engines can be understood as a combination of both SI and CI engines, operating without throttling. It implies that these types of engine admit as much air as it is practicable to drop into the cylinders. Natural gas is usually introduced and mixed with the air in the inlet manifold, so that EGR application involves displacement of some of the inlet charge air by the recirculated exhaust gas. The consequence of this displacement is a reduction of the air available for combustion, which leads to a parallel reduction of the air-fuel ratio. This reduction in AFR can significantly affect exhaust emissions [28].

Sometimes, EGR slightly raise the intake charge air temperature even with a cooling system installed, as it has been proved in this thesis, but, at the same time, cause lower combustion temperatures in the cylinder. It is due to the previous mentioned reduction of the AFR, as well as specific heat, which, accompanied by a shorter ignition delay, implies a decrease in the early part of combustion. Thus, thermal efficiency is improved with EGR, especially at higher natural gas percentages, as it was demonstrated by Abd-Alla in a dual-fuel engine at 2,000 rpm and for both ¼ and ½ loads [28].Thermal efficiencies at ¼ load are shown in figure 2.15.

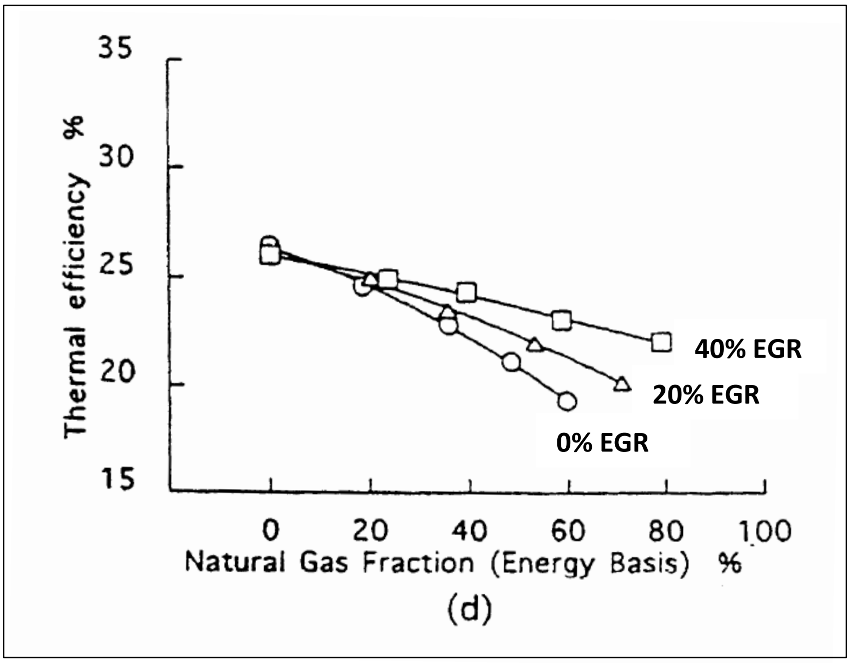

Figure 2.15: Thermal efficiency as function of natural gas fraction for different EGR rates in dual-fuel engine (2,000 rpm, ¼ load) [28].

3. Research equipment

3.1. Research engine

The engine used in this master's thesis is a single cylinder high-speed research engine based on a commercial six cylinder AGCO Power 84 CTA heavy duty engine.

The fuel is injected only into one of the six cylinders mentioned above, with one injection per working cycle. It uses the original commercial cylinder head and piston and the five empty cylinders have pistons only for balancing purposes, but they are depressurized with holes in them allowing air to flow freely, so no compression work is done.

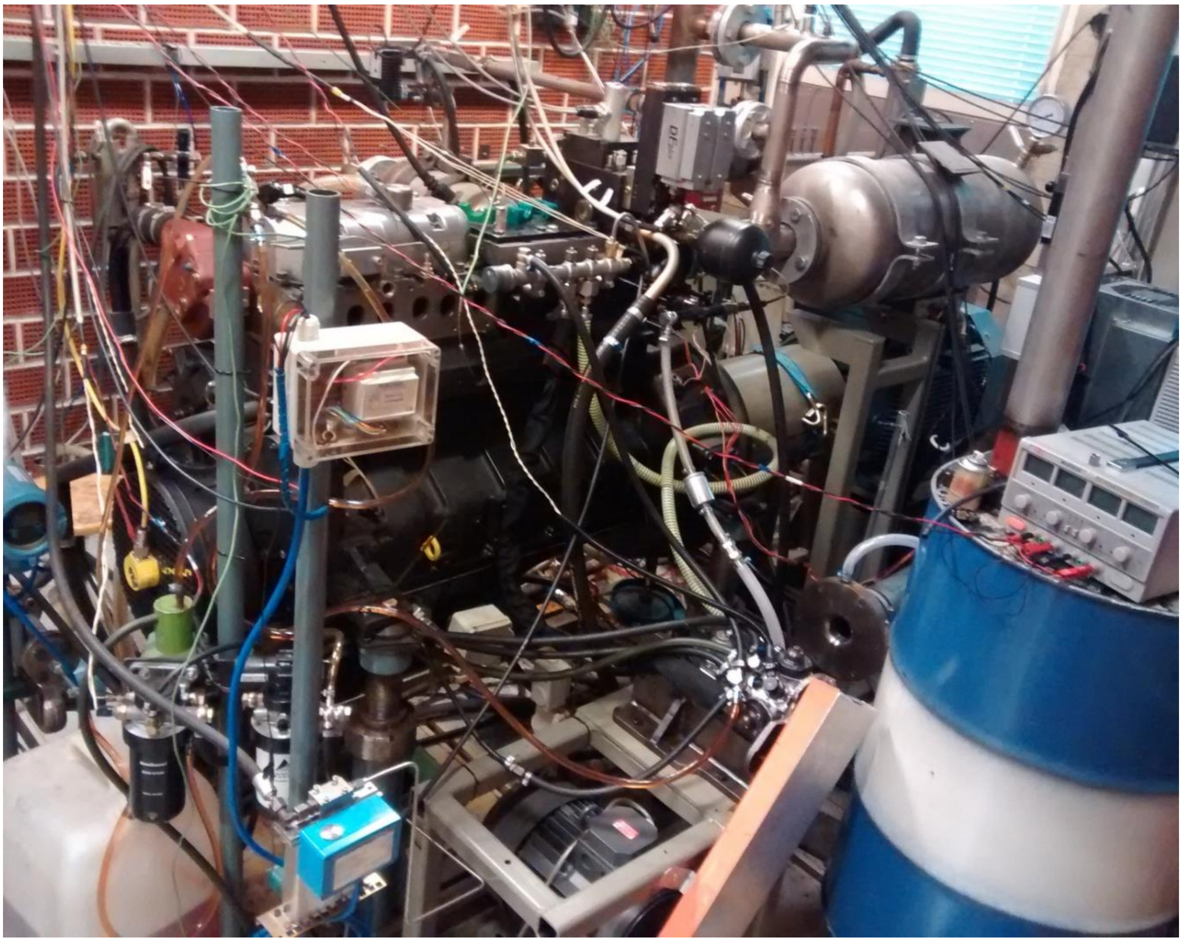

Figure 3.1: Single cylinder high-speed research engine.

Diesel injection is performed by a common rail direct fuel injection system, which allows the nozzle to have high injection pressures.

The engine uses an electric motor with a frequency converter as a dynamometer.

Due to the lack of a turbocharger installed, instead there is an external air compressor which supplies the charge air to the engine, being regulated by a controlled valve.

Exhaust gas pressure in the exhaust pipe is also controlled. The air pressure that can be achieved with this system is approximately 6 bar. Furthermore, the charge air system can also adjust the charge air temperature through a cooler and an air heater. In addition, the mass flow can be measured with a Micro Motion mass flow meter.

This engine is equipped with an electro hydraulic valve actuator system (EHVA), which allows us to adjust the gas exchange valve timings and use of different maximum lifts freely. It simulates the camshaft of a real engine.

Some modifications were made recently in the engine in order to get a dual-fuel engine. Apart from the diesel injector that was previously installed, a couple of CNG/LPG gas injectors have been added to the intake manifold, where methane is mixed with the air prior to arriving to the combustion chamber. The methane comes from a compressed gas bottle. An EGR valve has also been included in the exhaust pipe, as well as a pressure container to absorb the pulses of the exhaust gases, which is easily called exhaust pulse absorber. This manifold is used to absorb the pressure pulses just before the exhaust valve closes. Moreover, another similar manifold, gas feed pulse absorber, was added in the gas intake pipe in order to keep pressure within adequate values, due to some problems with the gas injection during the first dual-fuel tests.

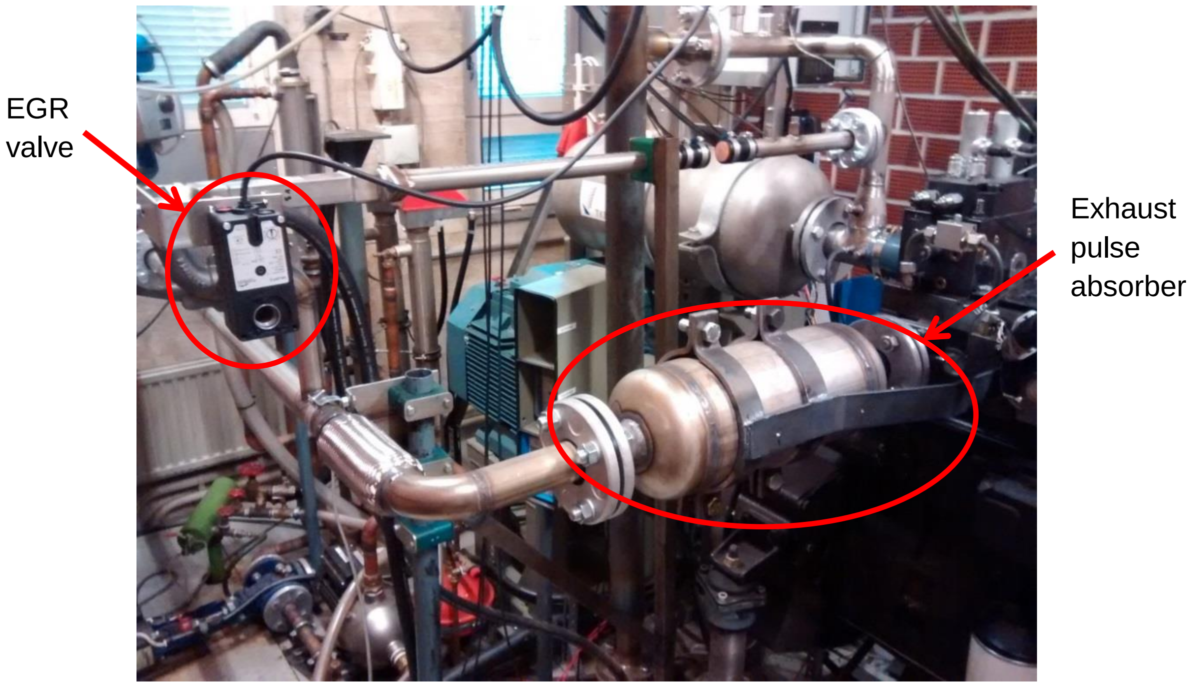

Figure 3.2: EGR valve and exhaust pulse absorber.

In addition, the different fuel injection parameters (injection timing, injection duration, injection pressure) can be varied freely.

Main engine specifications are reflected in the table 3.1.

Table 3.1: Research engine specifications.

Parameter	Value	Unit
Engine type	AGCO Power 84 CTA 4V	-
Number of cylinders	6 (1 in use)	-
Cylinder diameter	111	mm
Stroke	145	mm
Compression ratio	17:1	-
Fuel type	Dual-fuel (methane + diesel)	-

3.2. Control system

The engine is controlled from a PC, in which there is a software program developed and optimized by some previous Master´s thesis students. The main software development was made in 2009 through Matlab and Simulink codes, but several changes have been made over the last few years. Some new parameters were included in the Matlab-Simulink code in order to control the methane injection as well as the new EGR valve.

The control system is conducted by the real-time system dSPACE, based on a PowerPC processor powered computer. It is also called soft real-time system, which has an approximate duration of the control loop of 80 µs. However all the functions are conducted in the time frame of the fixed step-size.

This dSPACE real-time system is performed by a real-time computer, seen in figure 3.3, connected with input and output cards (figure 3.4), and a graphical user interface running on the PC. They are connected via Ethernet connection, and both the input-output card and the real-time computer are placed into a control cabin. In this cabin there are also some power sources and signal converters which are required for the measurement and control devices.

Figure 3.3: dSPACE real-time computer.

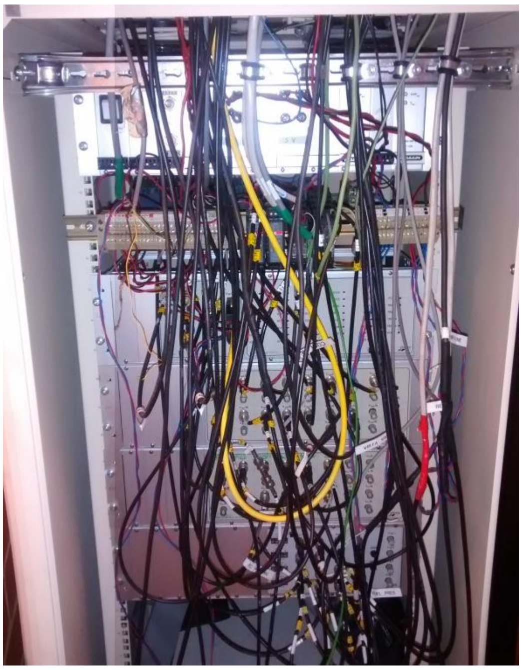

Figure 3.4: Inputs / outputs.

It is worth mentioning that the graphical user interface tool for dSPACE is ControlDesk, experiment software for research engine management. It performs all the necessary tasks, providing a single working environment, to use it for ECU measurement, calibration and diagnostics mainly.

For better understanding the control system is divided into different subsystems, each one controlling one part of the full system. A schematic overview of this system is shown in figure 3.5.

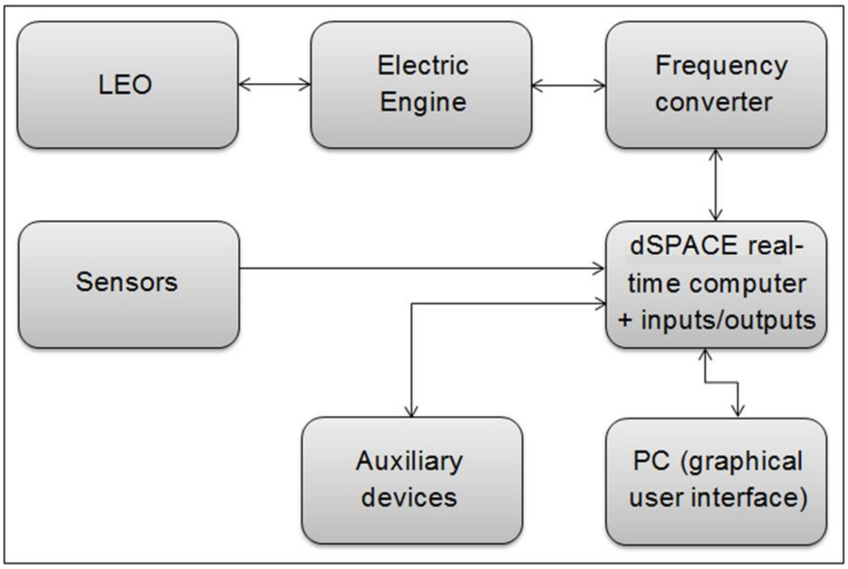

Figure 3.5: Schematic overview of the control system.

The control program consists in a set of input and output subsystems, and all inputs and outputs use voltage signals. The input subsystem includes interfaces with A/D converters and formats the input signal to conventional values of scaling and offset, while the output subsystem achieves the same backwards. It is divided into several subsystems which allow us to control different functions, such as charge air control or injection control.

Must take into account that this kind of software is very expensive and, in addition, it should be implemented by the user for specific purposes.

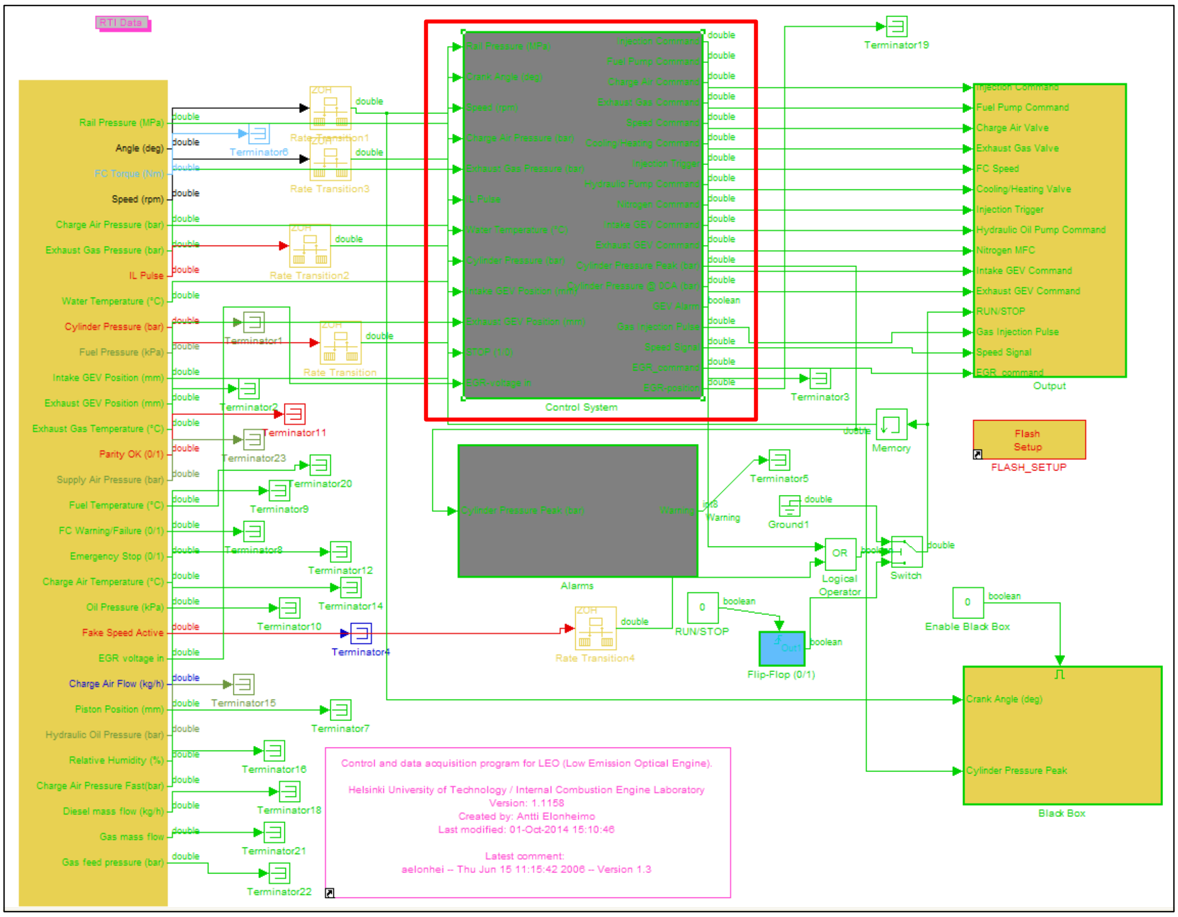

Figure 3.6: Control system of the research engine.

3.3. Measurement tools

3.3.1. Engine parameters measurement

In order to make all the required measurements for the project, the same software as for the control system was used, which was introduced some lines above, dSPACE.

With this real-time system, a capture of all measured data can be taken in Matlab file format, in order to be analyzed later. The tool creates 25.000 measurement data of each parameter during 2 seconds of engine running.

Engine´s data system is equipped with low and high frequency sensors that provide us the signals from the engine. In high frequency, the crank angle and cylinder pressure sensors are used, allowing maximum precision, being the rest of the sensors and other instruments low frequency sensors.

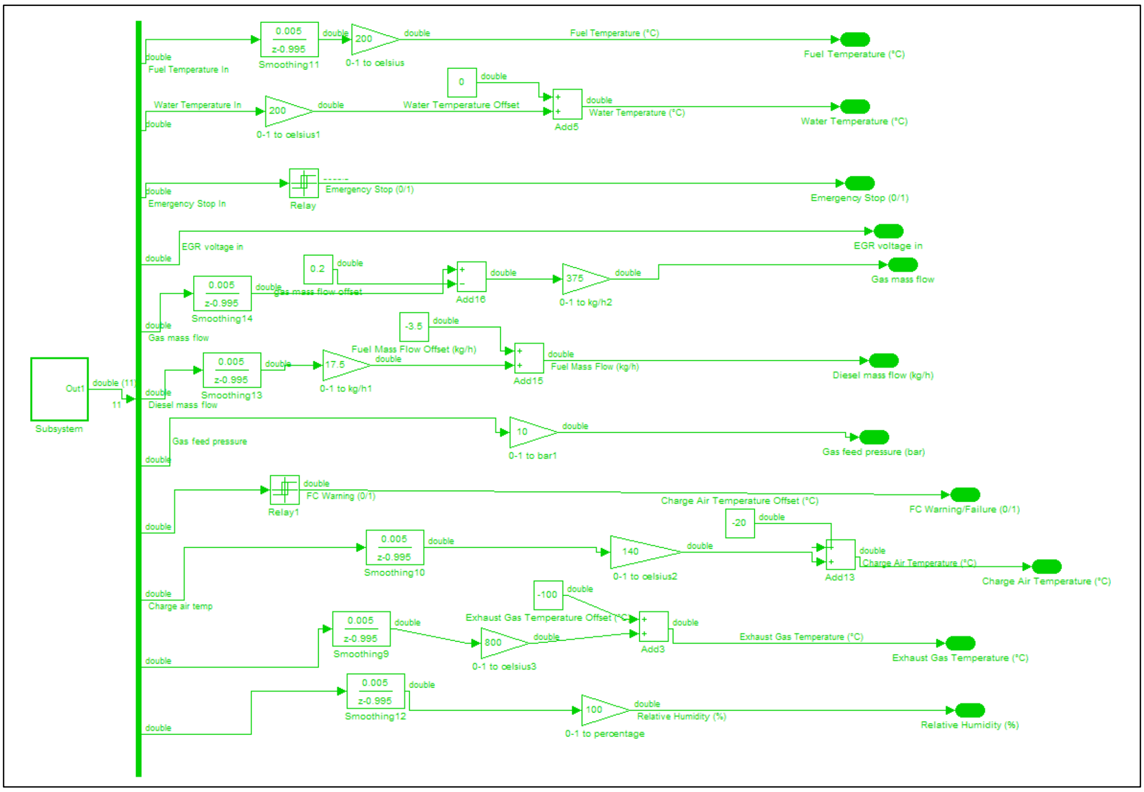

Figure 3.7: Inputs from sensors in Simulink.

3.3.2. Emissions measurement

Regarding to the emission measurement, other computer software was used. It is called LabVIEW, which provides the capacity of measure NO, NO_2, NO_x, HC, CO, CO_2 and O_2 emissions. The sample is taken just after the exhaust manifold and driven into the emission pre-filter, in order to take it then into different and separate analyzers, using for this a measurement frequency for emissions of 1 Hz too. Measurements in this thesis were taken calculating the average of the emission amounts during 30 seconds.

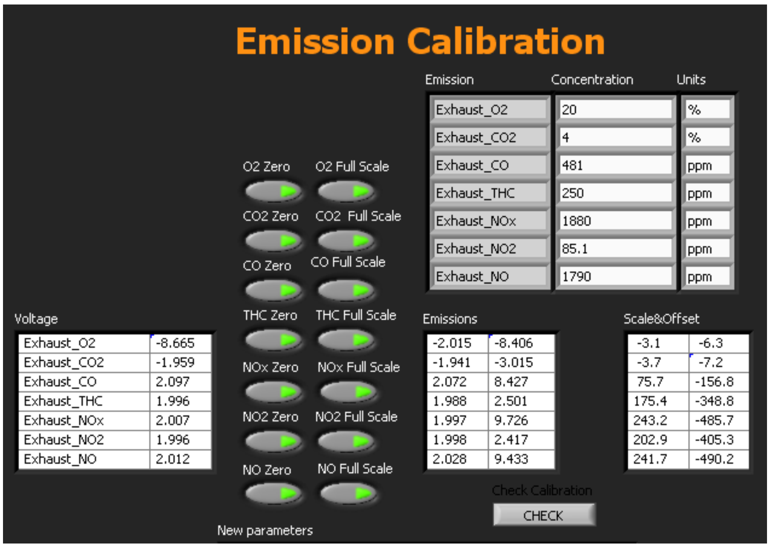

Figure 3.8: Emission calibration window.

In order to take the measurements for the LabVIEW interface, the device presented in figure 3.9 was employed. First of all it is calibrated with the different reference gases that should be analyzed (NO_x, HC, CO, O_2, CO_2) and then it is ready to use.

Each measurement day the emission device should be calibrate, checking the concentration of the bottles and writing them down in the calibration window (figure 3.8). Also zero and full scale values, as well as scales and offsets, should be compare with the ones of the previous days before saving, to make sure they are around the same ranges.

Figure 3.9: Emissions measurement device.

3.3.3. EGR rate measurement

This master's thesis is focused on the influence of the EGR parameter on the combustion and the emissions of a dual-fuel engine. Therefore, it was necessary to calculate this rate.

To achieve this goal, as it will be explain in point 3.4.1, CO_2 inlet value (intake air + recirculated exhaust gases) and CO_2 exhaust value were needed.

CO_2 exhaust value was obtained from LabVIEW. However, the inlet one could not be seen in the same program since no sensor was installed and connected to LabVIEW. For this reason, another sample was taken from the intake pipe, which value could be read in a different analyser that was set near the engine. It was easier than connect the sensor to the previous analyser, which was placed further. This device is seen in figure 3.10, where a CO_2 volume percent can be read.

Figure 3.10: Inlet CO_2 analyser.

3.4. Processing of results

Processing of all the experimental results from the measurements has been made basically with Matlab and Microsoft Excel computer programs.

Matlab has been used to calculate cylinder pressure and heat release curves due to its high efficiency and its powerful tools for processing matrix, structure and cell data. As Matlab is more sophisticated, it provides a better attenuation of the noise and makes the study simpler, since the accuracy is greater.

Analysis of the emissions, as well as other parameters such as lambdas or some temperatures, has been done through Microsoft Excel, as it is easier when the amount of data is not high.

In this section, some equations used during the study in both previous programs are presented. Besides, taking into account that experimental research has typically some sources of uncertainties, they are also considered here to be aware of them when analysis and conclusions are made.

3.4.1. Post-processing

Heat release

Cylinder pressure versus crank angle data over the compression and expansion strokes of the engine operating cycle can be used to obtain quantitative information on the progress of combustion, such as Heat Release Rate (HRR) curves.

Prior to heat release calculation, an averaged pressure curve from the different cycles was obtained, as well as the application of a low-pass filtering in order to make that curve smoother, eliminating most of the disturbance, and easier to understand. It is important since HRR is calculated as a derivative function.

Heywood [10] includes a demonstration of the heat release expression (3.1) assuming that the contents of the cylinder can be modelled as an ideal gas.

$$\frac{dQ_n}{dt} = \frac{\gamma}{\gamma - 1} p \frac{dV}{dt} + \frac{1}{\gamma - 1} V \frac{dp}{dt} \tag{3.1}$$

where Q_n is the net heat release rate (in J/^{o}CA)
γ is the isentropic constant (≈1,35)
p is the cylinder pressure as a function of crank angle (in Pa)
V is the cylinder volume as a function of crank angle (in m^3).

Derivative of cylinder pressure, as it is a discrete value function of crank angle, can be simplified as it is written below (3.2).

$$\frac{dp}{dt} = \frac{dp}{d\theta} = \frac{\Delta p}{\Delta\theta} = \frac{p_{i+1} - p_{i-1}}{\theta_{i+1} - \theta_{i-1}} \tag{3.2}$$

where θ is crank angle (^{o}CA).

Likewise, derivative of cylinder volume can be calculated as follow.

$$\frac{dV}{dt} = \frac{dV}{d\theta} = \frac{\Delta V}{\Delta\theta} = \frac{V_{i+1} - V_{i-1}}{\theta_{i+1} - \theta_{i-1}} \tag{3.3}$$

Lastly, the calculation of this cylinder volume at any crank angle position θ can be made with the next formula [10].

$$V = V_c + \frac{\pi B^2}{4}(l + a - s) \tag{3.4}$$

where B is the cylinder bore
l is the connection road length
a is the crank radius
s is the distance between the crank axis and the piston pin axis, and is given by

$$s = acos\theta + \sqrt{l^2 - a^2 sin^2\theta} \tag{3.5}$$

EGR rate

The percentage of exhaust gas recirculated (*EGR(%)*) is defined as the percent of the total intake mixture which is recycled. Thus,

$$EGR(\%) = \frac{\dot{m}_{EGR}}{\dot{m}_{air} + \dot{m}_{fuel} + \dot{m}_{EGR}} \times 100 \tag{3.6}$$

where $\dot{m}_{EGR}$, $\dot{m}_{air}$ and $\dot{m}_{fuel}$ are the mass flows of the exhaust gases recirculated, the air and the fuel (methane + diesel), respectively.

However, when several measurements are done consecutively, an efficient equation to calculate the EGR rate can be easily reached from a CO_2 balance in the inlet port.

$$[CO_2]_{atm} \times \dot{m}_{air} + [CO_2]_{EGR} \times \dot{m}_{EGR} = [CO_2]_{inlet} \times (\dot{m}_{air} + \dot{m}_{EGR}) \tag{3.7}$$

Using equations (3.6) and (3.7), considering that CO_2 concentration in the recirculated exhaust gases is the same as in the exhaust, and disregarding the CO_2 concentration of the atmospheric air, the formula used in this thesis can be obtained and seen in (3.8).

$$EGR(\%) = \frac{[CO_2]_{inlet}}{[CO_2]_{exhaust}} \tag{3.8}$$

Methane lambda

Lambda parameter, also known as air-fuel equivalence ratio, is the ratio between the real air-fuel ratio (AFR) and the stoichiometric one for a given mixture.

In dual-fuel case, two different lambda parameters should be taken into account. On one hand the relation between the amount air and methane, and on the other hand the one between the air and both fuels (methane and diesel).

It is also important to have in mind that some air is being recirculated to the intake manifold as oxygen percentage. This value is included in (3.10) as the mass flow of air recirculated ($\dot{m}_{air\ recirculated}$), calculated taking into account the EGR mass flow and the oxygen concentration in the recirculated exhaust gases.

EGR mass flow ($\dot{m}_{EGR}$) can be obtained from (3.6), being known the EGR rate (3.8) and the rest of the mass flows from the measurements.

Methane lambda is calculated as follow:

$$\lambda = \frac{AFR_{CH_4}}{AFR_{CH_4\ stoich}} \tag{3.9}$$

where $AFR_{CH_4\ stoich} = 17{,}2$ is the stoichiometric air-fuel ratio of methane
AFR_{CH_4} is the actual air-methane ratio, given by

$$AFR_{CH_4} = \frac{\dot{m}_{air} + \dot{m}_{air\ recirculated}}{\dot{m}_{CH_4}} \tag{3.10}$$

Dual-fuel lambda

For the calculation of this lambda, methane and diesel mass flows are used, as the ratio after diesel injection is searched.

Methane lambda takes more importance for the analysis, but checking if this one has a similar trend can show up if the calculation has been made correctly.

In this case, methane and diesel masses should be taken into account since both fuels are used for the calculus. Methane percentage is calculated with the following basic equation (3.11), using the averaged mass flows of both fuels for each measurement point.

$$m_{CH_4\%} = \frac{\dot{m}_{CH_4}}{\dot{m}_{CH_4} + \dot{m}_{DI}} \tag{3.11}$$

Dual-fuel lambda obtaining equation can be seen here:

$$\lambda_{DF} = \frac{AFR_{DF}}{AFR_{CH_4\ stoich} \times m_{CH_4\%} + AFR_{DI\ stoich} \times (1 - m_{CH_4\%})} \tag{3.12}$$

where $AFR_{CH_4\,stoich} = 17{,}2$ is the stoichiometric air-fuel ratio of methane
$AFR_{DI\,stoich} = 14{,}6$ is the stoichiometric air-fuel ratio of diesel
AFR_{DF} is the actual air-fuel ratio of the methane-diesel mixture, given by

$$AFR_{DF} = \frac{\dot{m}_{air} + \dot{m}_{air\ recirculated}}{\dot{m}_{CH_4} + \dot{m}_{DI}} \tag{3.13}$$

3.4.2. Uncertainties during experiments

Different problems were found during the realization of the needed measurements for this thesis, creating some uncertainties on the obtained results.

First unexpected incident was that dSPACE software was not working properly as to the measurement tool, disallowing data collection for several test points. Because of this problem, only engine measurement data was taken for full load and 40% gas fraction, with the exception of some values taken by hand and, of course, all the emission information. As well as this malfunction, some temperature sensors were not receiving the signal, so both charge air temperatures and exhaust gas temperatures were checked manually.

It is known that emission stabilization in this kind of research engine is slow, requiring long periods of time between measures for accurate results. Measurements in this thesis were taken in periods of 7-8 minutes, in order to avoid extremely high consumes of fuel and, thus, expending excessive amounts of money. In addition, only one measure was taken for each point. For these reasons, emission data could not be accurate enough in this thesis, and singular results can appear in some cases.

Some other deviations could be due to charge air pressure and exhaust gas back pressure fluctuations during the tests. Both at half and full load, pressures were establish to be constant; however they were slightly varying between cycles. In the same way, measurements were taken along different days, which can cause some data inaccuracy.

Lastly, it should be taken into consideration that CO emissions at 50% load could not be recorded because the CO ppm range of the emission measurement equipment was not wide enough to detect the values. Likewise, in this study 0% EGR rate was not possible to achieve whit the EGR valve absolutely closed, which should be due to some leaking in this aforementioned valve.

4. Analysis and discussion of results

4.1. Test program

Tests in this master's thesis were done in order to study the influence of the variation of the EGR rate parameter at 50% load and 100% load on dual-fuel methane combustion.

Results of the measurements with 100% load are more interesting, since the engine is working to the maximum. All measurement data used and obtained during tests, for both full and half loads, are shown in appendix A1. However, inasmuch as only all the parameters have been measured for an amount of gas of 40% and full load, the study is focused in the six points shown in table 4.1, where pre-establish data and measured data have been separated by a thicker line. The rest of the points are explained based on these results and some known experiments of other authors, considering, of course, all data taken by hand and emission results. This is the reason why full load operation was firstly studied, so half load analysis could be better done at a later stage.

Table 4.1: Measurement points at 100% load and 40% gas percentage.

Point	%Gas	Gas injection duration (ms)	Diesel injection duration (μs)	SOI (°CA)	Charge air pressure (bar)	Exhaust gas back pressure (bar)	EGR valve opening (%)	Charge air temp. (°C)	Exhaust temp. (°C)	EGR rate (%)
42	40	8	620	- 5	2.55	2.70	0	28	460	3
43	40	8	620	-5	2.55	2.70	5	31	460	6
44	40	8	620	-5	2.55	2.70	10	38	460	10
45	40	8	620	-5	2.55	2.70	15	50	488	17
46	40	8	620	-5	2.55	2.70	20	68	520	24
47	40	8	620	-5	2.55	2.70	25	90	540	29

All the matrices with the 41 previous other points can be seen in appendix A1. As seen, methane percentages are increased from 0% to 80% for half load, and from 0% to 40% for full load. The amount of gas could not be higher for this load due to its difficulty of stabilizing the cylinder pressure. Peak pressures over 250 bar were reached. Moreover, injection timing had to be modified in order to get these peak pressures below this value, so that SOI was 14°CA BTDC when 80% CH_4 was used at half load, instead of 7°CA BTDC as the rest of the previous test points. It is clearly specified in appendix A1.1.

At 50% load, measurements with 90% amount of gas have not been represented in this thesis, even though emission data and other parameters were recorded (appendix A1). Even observing adequate cylinder pressure values with earlier SOI during tests, performance is not clear after the analysis and no enough data were available in order to explain it, such as lambdas, cylinder pressures, HRR or HR.

In each of these natural gas percentages, EGR valve was opened from 0% to 25%, achieving EGR rates of around 30%, according to equation (3.8).

Several data was taken into account in order to discuss the results. Thus, charge air temperatures, exhaust gas temperatures, lambdas, cylinder pressures, heat release rates, cumulative heat release rates and NO_x, THC, CO emissions are presented below during the analysis.

4.2. Operation at 100% Load

As previously said, three different gas percentages were performed at full load, 0%, 20% and 40%. The first one is equivalent to the diesel mode, which is basically a reference for the emissions performance on dual-fuel combustion. Behaviors with 20% and 40% methane percentages are highly similar for all the emissions. Therefore, it was not a major issue, in this case, to have only the 40% gas data for the analysis, since explanations can work for both.

4.2.1. Engine operating temperatures

Seventeen points were taken at full load, varying natural gas percentage from 0% to 40% as said before. The last six points (42-47), represented in the graphs by green color, constitute different EGR rates for 40% gas, and, as mentioned in the previous chapter, these are the test points that were used for the whole parameter analysis.

Charge air and exhaust gas temperature values were taken by hand. Besides, in these charts it can be seen that the last point for 20% gas is not included, with a 25% of EGR valve opened, since measurement data was not recorded in that moment, but tendencies are perfectly clear anyway. Figure 4.1 represents exhaust gas temperature variation as function of the EGR rate.

It is interesting that exhaust gas temperatures rise when starting to increase methane percentage to 20%, but then it goes down when arriving to 40%. It is a peculiar phenomenon; although the values range is small and it is difficult to draw clear conclusions from this behavior. Taking this into account, importance was given to increasing trends more than to the actual values.

Similar phenomenon can be perceptible for charge air temperatures in figure 4.2, although numbers are lower so differences are smaller too.

Moreover, EGR tends to slow down the combustion and thus a bigger portion of the fuel burns later in the cycle, which could mean one of the reasons of this exhaust gas temperatures raise.

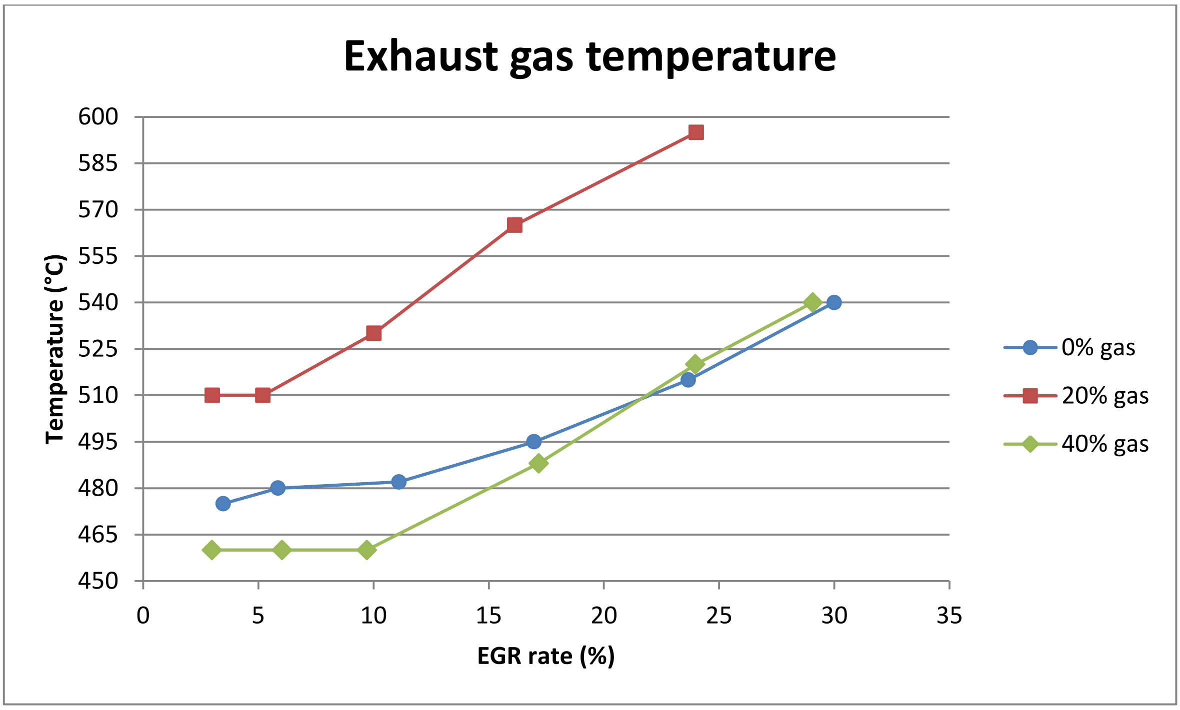

Figure 4.1: Exhaust gas temperatures as function of EGR rate at 100% load for different gas percentages.

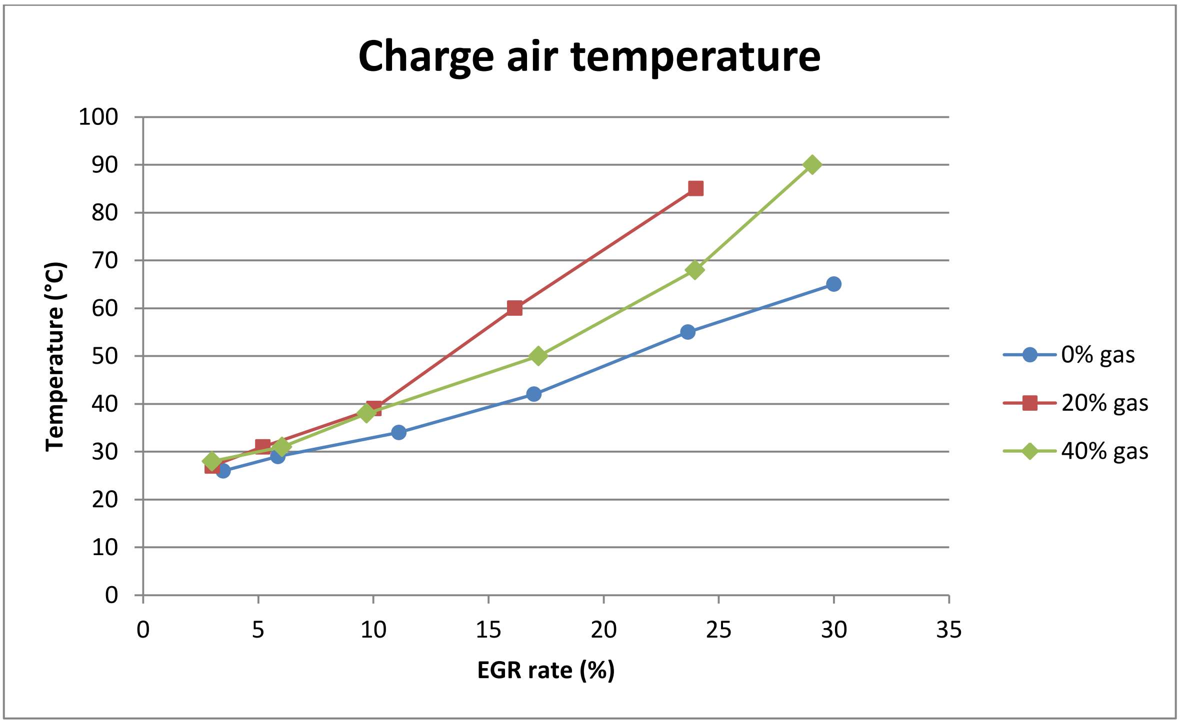

Figure 4.2: Charge air temperatures as function of EGR rate at 100% load for different gas percentages.

Similarly to half load curves, steeper slopes for charge air temperature can be noticeable when increasing the gas percentage, especially with 40% and higher EGR rates.

It is important to mention that, evidently, these temperature samples have been got with a sensor placed in the intake pipe after the recirculation is done. Charge air temperatures prior to recirculation keep practically constant.

Next chapters (4.2.2 - 4.2.5) only include results for the six previous mentioned points, with fuel amounts of 40% methane and 60% diesel, which measurement matrix was advanced in table 4.1.

4.2.2. Methane lambda and dual-fuel lambda

Lambda parameter study is very important to determine some behaviors on the emissions performance. As seen in chapter 3.4.1, it represents the relation between the actual air-fuel ratio of the mixture and the stoichiometric one.

When lambda exceeds the value of 1 the mixture is lean, which can imply difficulties in ignition and misfire could start to occur. On the other hand, maximum power is produced for rich mixtures ($\lambda<1$), compromising the emissions then.

On dual-fuel mode, methane lambda, for the premixed natural gas and air before entering the cylinders, is supposed to be around 2.2 at most according to Wärtsilä experiments [30] on lean-burn dual-fuel combustion, value above which misfiring could appear.

In this master´s thesis, both methane lambda, from the air-methane premixture, and dual-fuel lambda, after diesel injection, have been calculated.

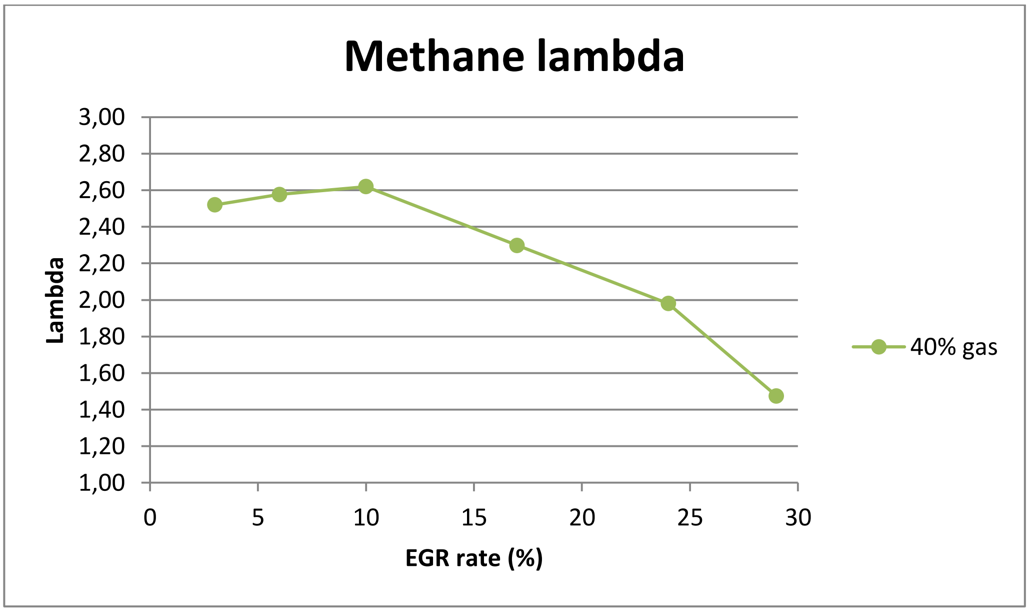

Figure 4.3: Methane lambda as function of EGR rate at 100% load and 40% gas.

It is interesting to compare both lambdas, since they follow the same trend, so it gives a confirmation of the correct calculation done. It also has sense that the mixture becomes richer when increasing the EGR rate, as exhaust gas recirculation implies a reduction of the oxygen concentration.

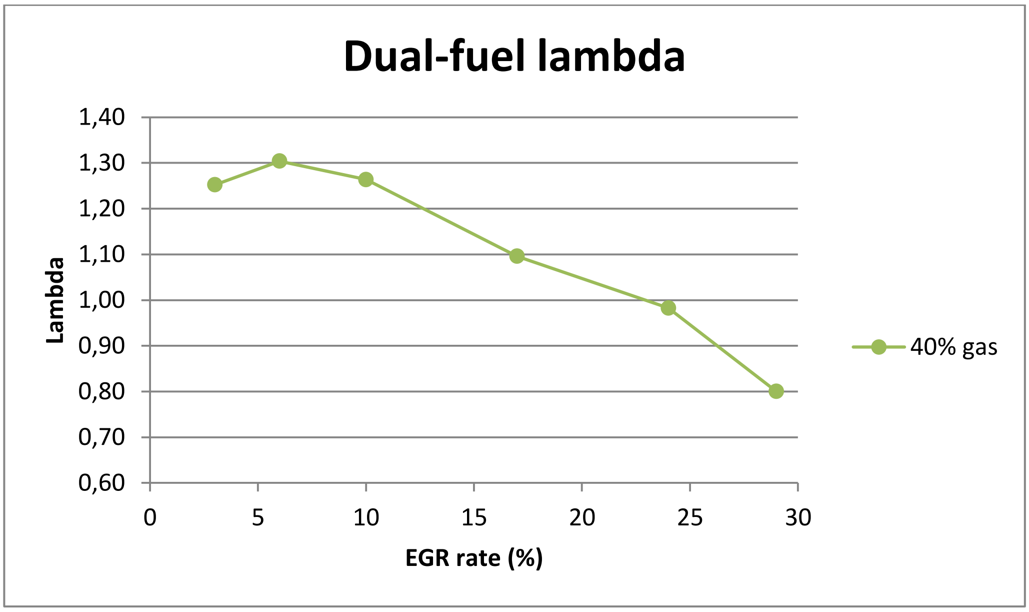

Figure 4.4: Dual-fuel lambda as function of EGR rate at 100% load and 40% gas.

4.2.3. Cylinder pressure

Cylinder pressure per crank angle degree curves for the six different points have been plotted in the same graph, in order to appreciate the differences when the EGR rate is raised. Also various colors have been used to differentiate them, although in some cases the lines are overlapped and they are difficult to distinguish, which means there was not a big difference in performance between those points.

Its calculation has been done taking the average of the data from 24 different cycles completed during 2 seconds of engine operation, obtaining more accurate curves than they would be with only one cycle.

Tendencies are not linear, but a thorough analysis of the graph can give some information. It is clear that peak pressure occurs later when increasing the EGR rate to 10%. However, when 17% rate is exceeded this peak starts to occur earlier in comparison with 3% EGR.

Slopes also become less steeper for the first rises of EGR rate, but then it returns to the same tendency of the beginning, giving evidence of a slower combustion for the first EGR rates and, maybe, faster for the highest ones. It can be confirmed by HRR and HR curves.

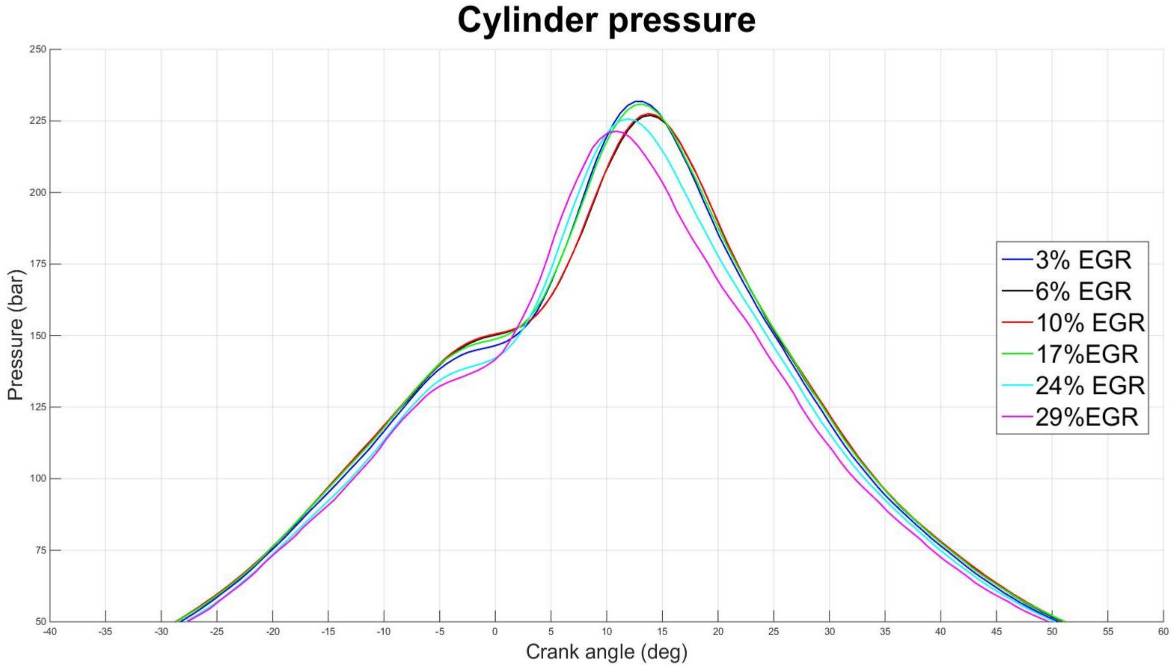

Figure 4.5: Cylinder pressure at 100% load and 40% gas for different EGR rates.

Finally, because of those reasons, it can be noticed from the chart that combustion starts earlier for both 24% and 29% natural gas percentages compared to the rest. SOI was the same for the 6 points (5 °CA BTDC), thus ignition delay seems to be decreasing for these two points mentioned above. This can be better confirmed from heat release rate curves too.

4.2.4. Heat release rate

Heat release rate (HRR) is the rate of heat produced by combustion in the engine. It means that the HRR curve starts rising at the same moment the combustion begins. However, when plotting a real heat release graph, several fluctuations can be found, and data have to be filtered in order to try to avoid this noise as much as possible.

HRR calculation in this thesis has used a low-pass filtering, even though some fluctuations are still persistent at the beginning and end of combustion. A low-pass filter uses a cutoff frequency. The filter passes signals with a lower frequency (passband), while attenuates those which are higher than the cutoff frequency (stopband). The attenuation amount depends on the filter design.

As discussed above, HRR curves should clarify the difference in ignition delays when varying the EGR rate. In fact, here it can be clearly seen that start of combustion for higher EGR rates happens slightly earlier. Duration of combustion can also be analysed from this chart.

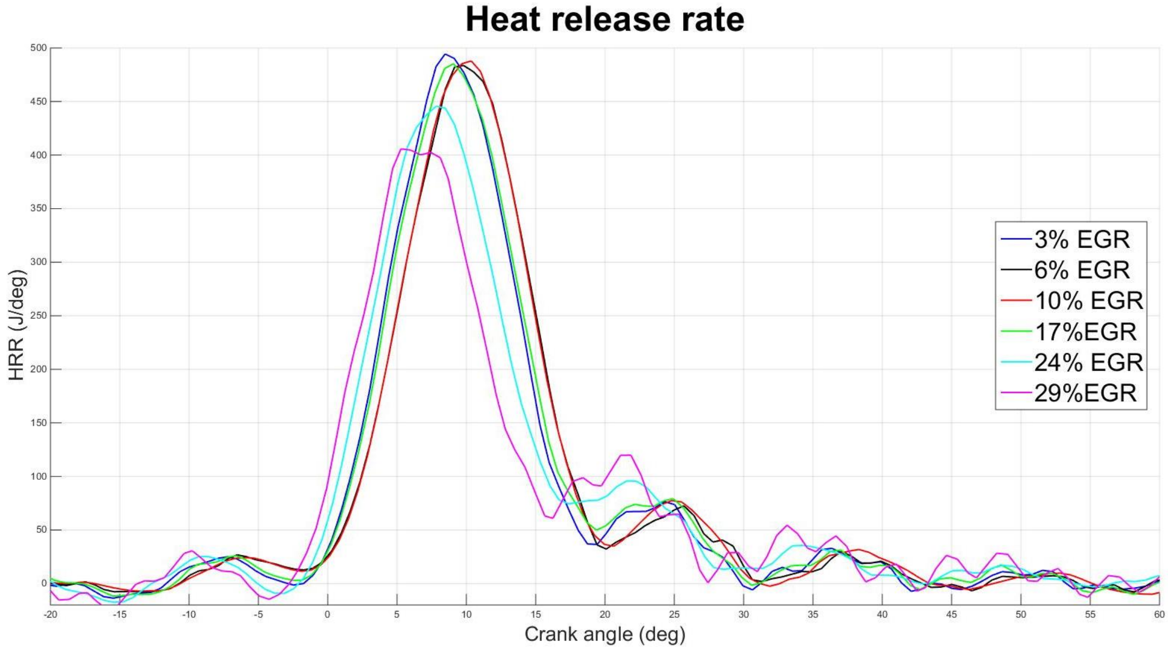

Figure 4.6: Heat release rate at 100% load and 40% gas for different EGR rates.

It is remarkable that curves are following similar trends as cylinder pressures did, firstly peaks happen later and then earlier than in 3% EGR case. At the same time, heat release peaks are reduced while EGR rate increases, especially for the highest EGR rates, when the decline is clearly and better seen. However, a second small peak can be better appreciated when EGR rates reach the highest values. It means there is probably some unburned fuel during the expansion phase, which burns very late. Therefore, combustion is not faster for highest EGR rates, but it is even slower, although it started to happen before.

4.2.5. Cumulative heat release rate

Finally, cumulative heat release rate (HR) has been calculated from the actual heat release values. This HR is an indicator of the combustion efficiency, since it shows the total fuel burned each time instant. The more the HR values grow, the more efficient the combustion is supposed to be.

As it can be deduced from the HRR chart, combustion seems to be less efficient for 24% and 29% EGR rates, at the same time that it happens faster and earlier. HR curves for the rest of the points follow approximately the same trend in terms of combustion efficiency. However, a clear shift can be appreciated in the start of combustion. It confirms the earlier combustion for higher EGR rates, as it was discussed based on HRR analysis.

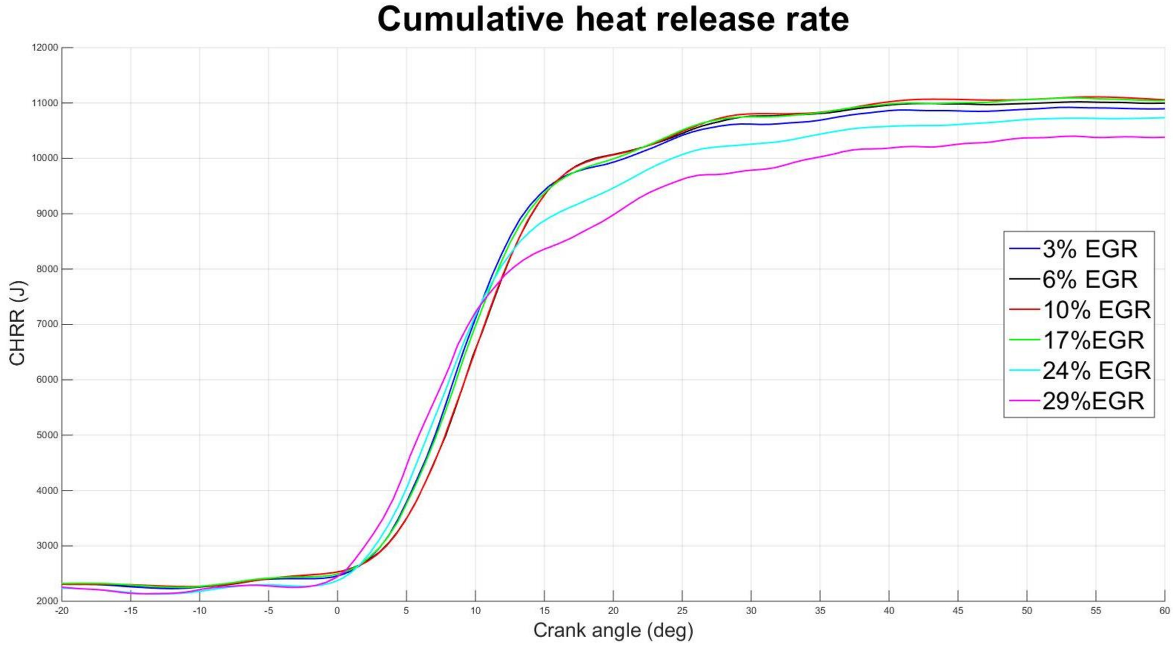

Figure 4.7: Cumulative heat release rate at 100% load and 40% gas for different EGR rates.

In the same way, these curves confirm that the end of the combustion is very slow for EGR rates of 24 and 29%. This performance should increase the exhaust gas temperature, so it is the one of the explanations of the temperature raise in the exhaust.

4.2.6. Emissions

No specific emissions (g/kWh) have been calculated, as basically trends while increasing EGR rate and methane percentage were interesting in this master's thesis.

Nitrogen oxides

As known, effect of EGR should cause a decrement in NO_x emissions. This common phenomenon can be confirmed for dual-fuel combustion as can be seen at both half and full load. Figure 4.8 shows this NO_x reduction at 100% load.

Trends of these curves are, thus, absolutely clear with a simple glance. NO_x emissions are being reduced as the EGR rate is increased for all the gas percentages. It is also noticeable that NO_x emission amounts during basic diesel combustion are higher in every point than the ones with dual-fuel mode.

EGR rate variation effect on NO_x emissions is totally evident and clear, given that one of the main functions of the exhaust gas recirculation is reducing them. Its reason is the incorporation of inert gases to the combustion, acting as an absorbent of combustion heat to reduce the peak in cylinder temperatures and pressures. It can be checked from figures

4.5 and 4.6. This decrease is even more important for full load, with a reduction percentage of more than 80%.

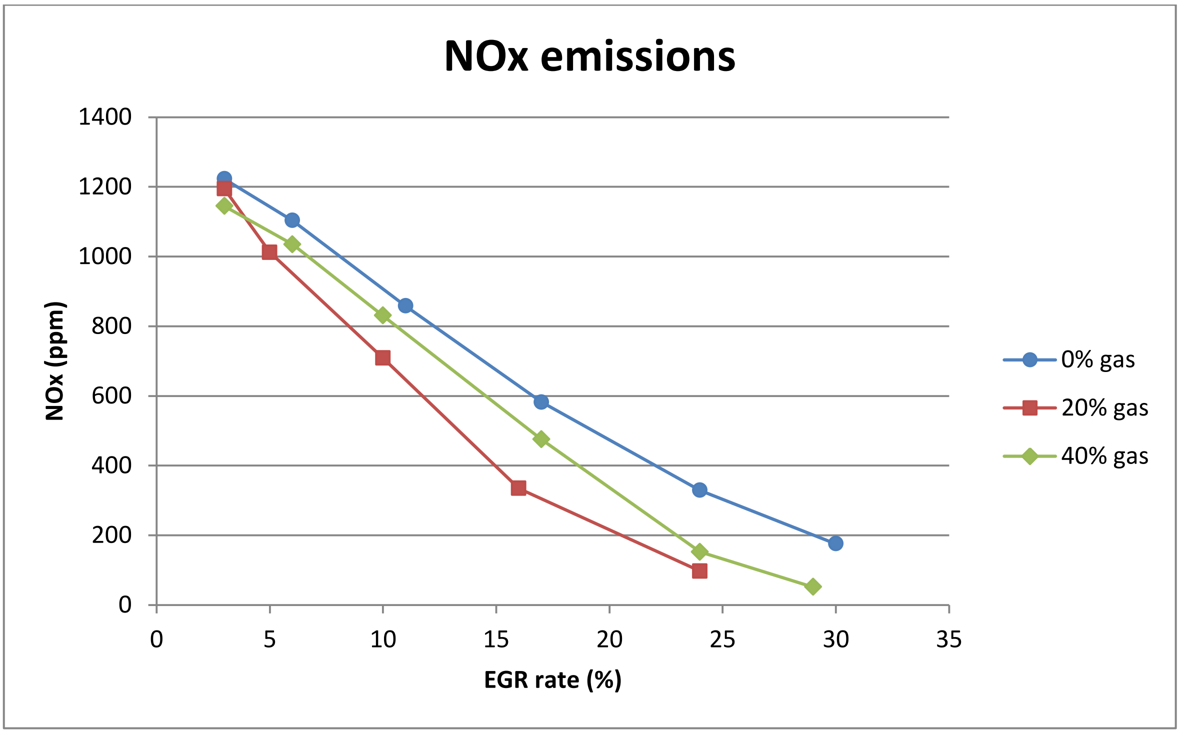

Figure 4.8: NO_x emissions as function of EGR rate at 100% load for different gas percentages.

Increases on charge air temperatures take an important part on NO_x emissions, usually increasing them, and lower proportions of EGR can be reached when these temperatures are not kept constant. However, EGR effect on these tests was intense enough to soften intake temperatures effect.

It can be also observed a change in NO_x curves slope for the highest EGR rates for all the natural gas percentages, which can be explained by the worse combustion efficiency found in the 40% gas analysis for the last two points, 24% and 29% EGR rates, but this phenomenon does not have much importance. Thus, no CH_4 percentage effect can be easily established with actual data on NO_x emissions performance at full load beyond its clear reduction when changing to dual-fuel mode compared to diesel.

Total unburned hydrocarbons

THC emissions performance when increasing EGR rate is not obvious, and its effect on dual-fuel combustion is not well known. Figure 4.9 shows its performance on measurements done in this thesis.

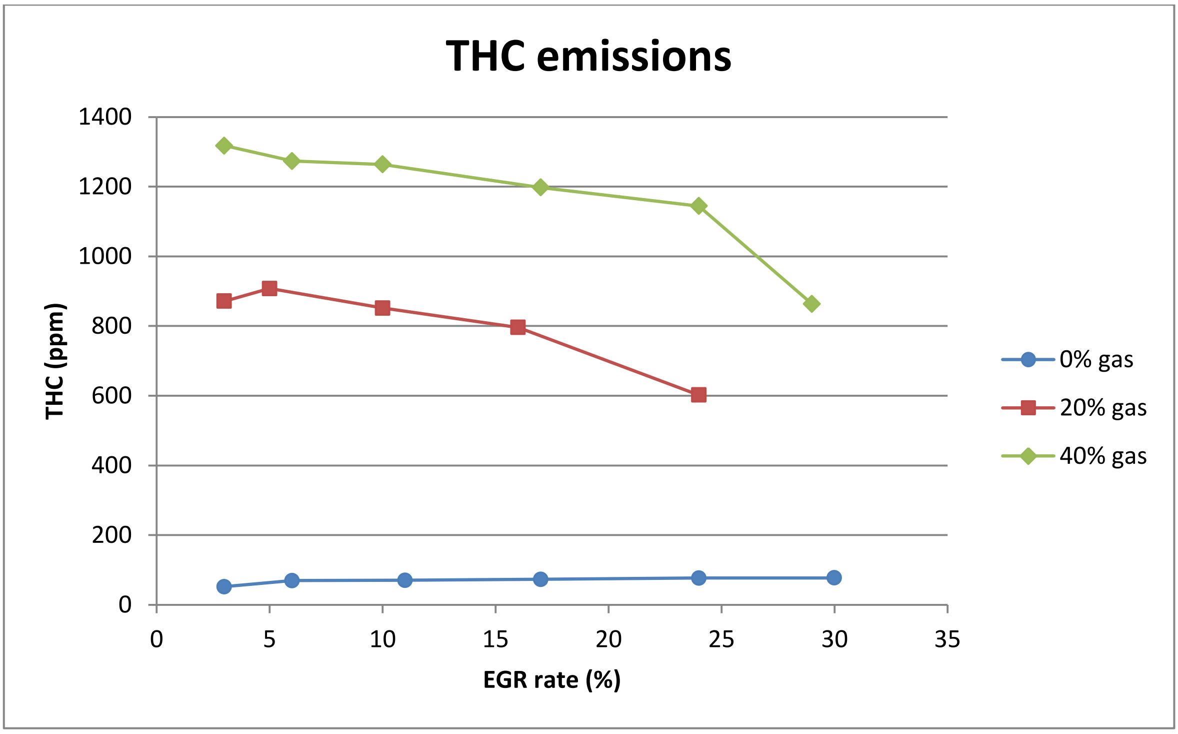

Figure 4.9: THC emissions as function of EGR rate at 100% load for different gas percentages.

Regarding to THC emissions at full load, it can be seen that reduction starts to be notable even with only 20% of methane, in contradistinction to half load, where changes are only appreciable from 60% natural gas percentage. Tendencies are similar in both cases, although the values range is lower and differences between diesel and dual-fuel mode become smaller too, being maybe the slopes slightly less steep at 100% load.

THC emissions are practically constant in diesel mode when EGR rate is raised, with a minute growth. However, apart from the range increment in THC emissions when increasing the CH_4 percentage and changing to dual-fuel operation, a notable change can be also seen from diesel mode, THC decrease with EGR rate.

THC raise under dual-fuel operation compared to diesel mode could be mainly due to the lower charge air temperature and air-fuel ratio, resulting in slower combustion and allowing small quantities of fuel to escape the combustion process, as well as the appearance of methane slip that is not present in diesel.

Decrement of THC emissions goes in the completely opposite direction that it is supposed to be in diesel mode. This behavior can be observed from the values of THC ppm when no methane is used, in smaller scale for a better appreciation (figure 4.10). It can be observed that THC rise slowly and in small amounts, as it was also mentioned and checked by Söderena, P. [18] in his thesis, where same engine in diesel mode was used.

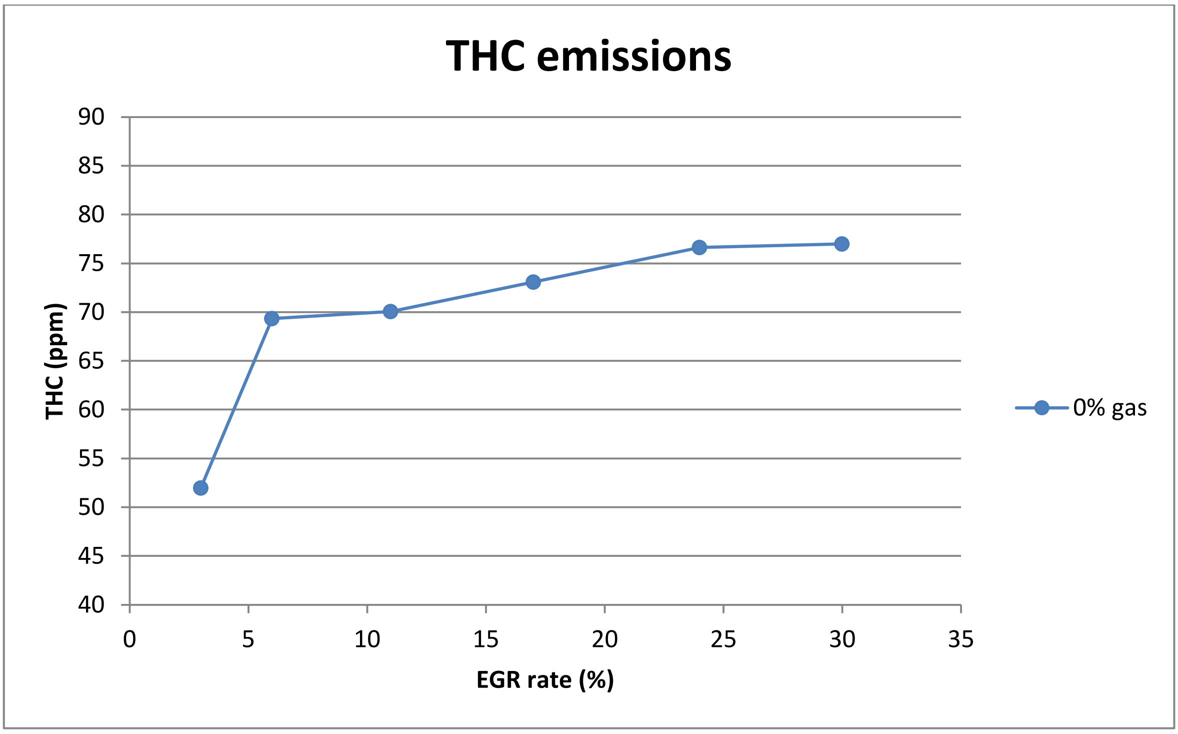

Figure 4.10: THC emissions at 100% load and 0% natural gas percentage

Focusing on dual-fuel mode performance regarding to THC emissions, some reasons have been found to explain this drop.

Firstly, higher amounts of unburned hydrocarbons at lower EGR rates can be related to lambda values. Looking at figure 4.3, extremely high methane lambda values are observed for 3, 6 and 10% EGR rates. These values are higher than 2.2, maximum lambda established by Wärtsilä [30] in their dual-fuel experiments above which misfiring could appear. Therefore, these high THC emissions at the beginning could be explained by excessive amounts of methane slip, since it is probable that this separation from the stoichiometric lambda implies incomplete combustion. Königsson [16], in his doctoral thesis performed on a dual-fuel engine, showed that HC emissions were decreasing with lower lambdas.

It is also interesting that some optical experiments done by Dronniou et al. [31] on a single-cylinder optical research engine in dual-fuel CNG/Diesel mode, showed that combustion when diesel injection was done started in the periphery of the cylinder and methane was being displace to its center and being progressively burned from outside inwards. Thus, when ignition delay is higher, especially for larger amounts of gas, as it happens here for lower EGR rates as seen in figures 4.5 and 4.6, higher amounts of methane slip could remain after the combustion around the center of the cylinder. Of course all of these are assumptions, but it could be a reason to explain the engine performance during these experiments, and provides some references to future work.

One reason for the constant decrement of THC emissions while increasing the EGR rate can be that the presence of radicals in the recirculated exhaust gases can help to initiate the combustion process, even more when increasing the charge air temperature due to mixing with exhaust gases [32].

Steeper decrease of THC emissions with higher EGR rates, especially for last point with 29%, can be due to different reasons. As said in the previous paragraph, an increase in the charge air temperature can help to this tendency, and it is clearly noticeable in figure 4.2 that the intake temperature slopes become steeper while EGR rate is raised. On the other hand, also shorter ignition delays and slightly slower combustions have been found for these highest EGR rates, which, as said before, seem to be a cause of the ability in which higher amounts of methane can be burned. However, it does not mean that combustion efficiency improves, as it is verified in the HR curves and even in CO emissions levels.

Carbon monoxide

Also CO emission measurements were got for this load, so its performance and relations with hydrocarbon trends can be studied.

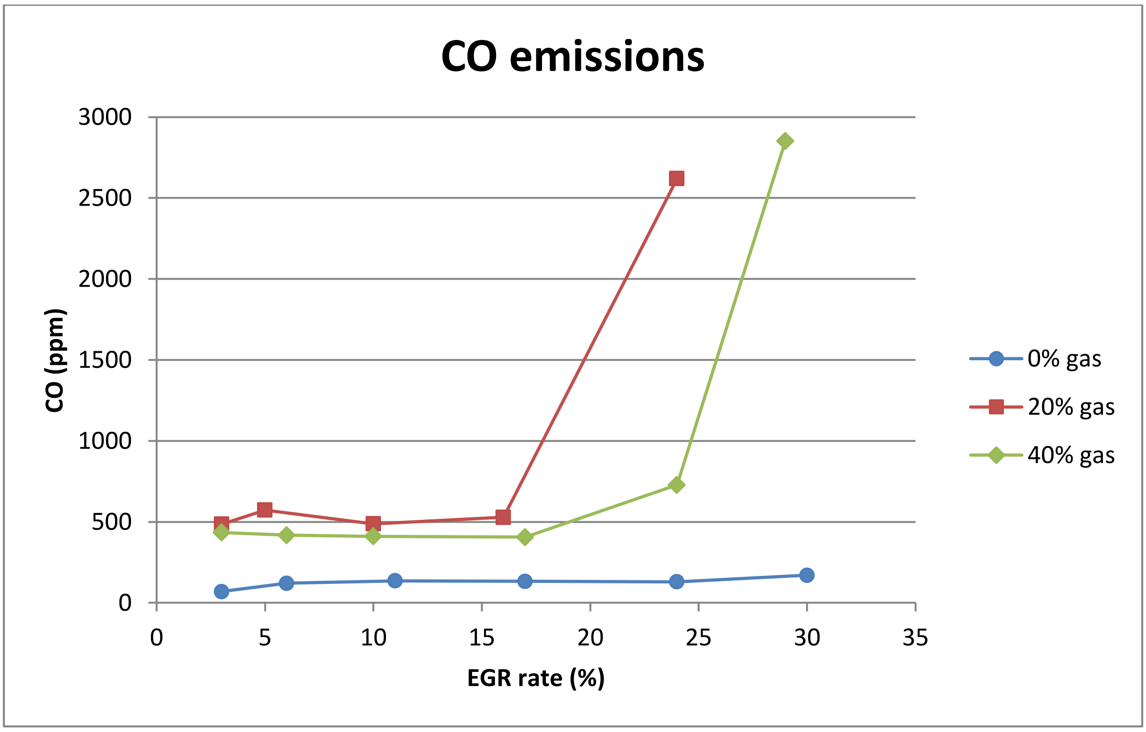

Figure 4.11: CO emissions as function of EGR rate at 100% load for different gas percentages.

As commented for THC, CO emissions are also higher for dual-fuel mode as it can be observed from figure 4.11. However, for the lowest EGR rates, not very high increase is noticed when injecting gas, so establishing a relation with THC emissions, it could be deduced that CH_4 percentage influence is also higher for lower loads.

On the other hand, the rise of CO emissions for the highest EGR rates is enormous, especially at 40% natural gas percent. Reasons can be searched in lambdas variation, as the mixture becomes richer with EGR, and also in the combustion efficiency, since high CO levels are usually a good indicator of combustion inefficiency.

It is known that in most of the cases CO and THC emissions go hand in hand. Nevertheless, it did not happen in these measurements, as it is perfectly seen in figure 4.12. Although CO emissions seem to be practically constant with EGR variation, an extremely sudden increase can be noticed for the highest EGR rates. This rise proceeds in the totally opposite direction as it does in THC emissions, where the abrupt change is in decreasing.

First and most important conclusion that can be extracted from CO emissions is the combustion efficiency drop for higher EGR rates around 25-30%. Usually high CO emissions mean less efficient combustion and, in these measurements, it can be confirmed from HRR and HR curves. Thus, it can be concluded that on dual-fuel mode extremely high EGR rates can cause losses of combustion efficiency.

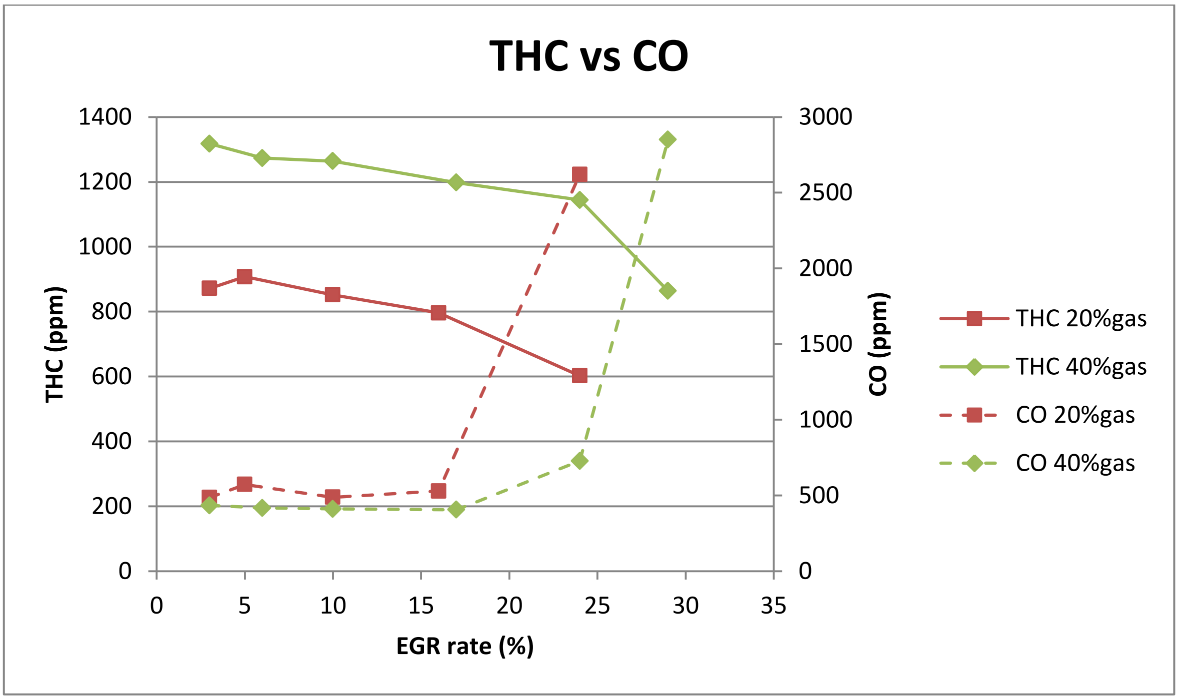

Figure 4.12: THC versus CO emissions as function of EGR rate at 100% load.

These sudden CO growths make sense looking at the lambdas variation, in figure 4.4. Dual-fuel lambda, for 40% gas percentage case, becomes lower than the stoichiometric for the last two points (24-29% EGR rate), turning the mixture into a rich one, exactly when CO levels start to rise. Therefore, higher CO emissions seem to indicate a fuel mixture richer than ideal in this case, accompanied by this fall in combustion efficiency. Also increases in charge air temperatures take part in this growing CO rise with EGR rate.

To sum up, at full load, NO_x emissions clearly decrease with EGR rate as supposed. However, dual-fuel mode in this research engine generates a contrary behavior in CO and THC emissions. Hydrocarbons seem to decrease due to the formation of a better mixture when increasing EGR rates, even with a reduction in the combustion efficiency, while carbon monoxide rises because of the decrement of oxygen concentration and, thus, excessively lower lambda and rich mixture.

4.3. Operation at 50% Load

4.3.1. Engine operating temperatures

As said before, measurements at 50% load have only recorded the emissions data. However, some essential operating parameters were taken by hand as they take importance for the later analysis of results, so that tendencies of charge air temperatures and exhaust gas temperatures can be analyzed here.

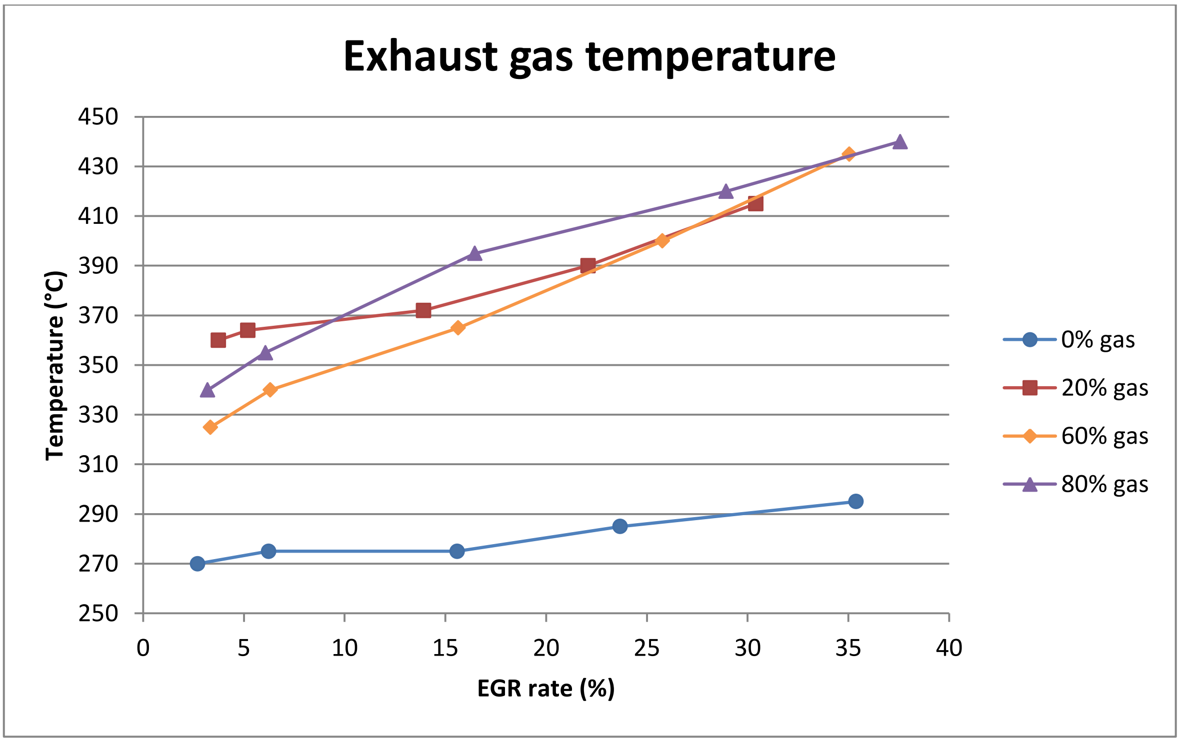

Figure 4.13: Exhaust gas temperatures as function of EGR rate at 50% load for different gas percentages.

Not all the points are included for a better understanding. Exhaust gas temperatures for 40% CH_4 percentage have been removed because there was an overlap with the 20%

gas curve, as well as 40% and 60% gas curves for charge air temperature for same reason. It is easily observed from figure 4.14 that even 20% and 80% charge air temperatures evolution is significantly similar.

Figure 4.13 shows that exhaust gas temperatures rise drastically when changing to dual-fuel mode. The evolution with the increase of the natural gas amount is not clear, but there is a clear ascent together with the EGR rate increment. It makes sense since the heating values of methane are higher than diesel ones.

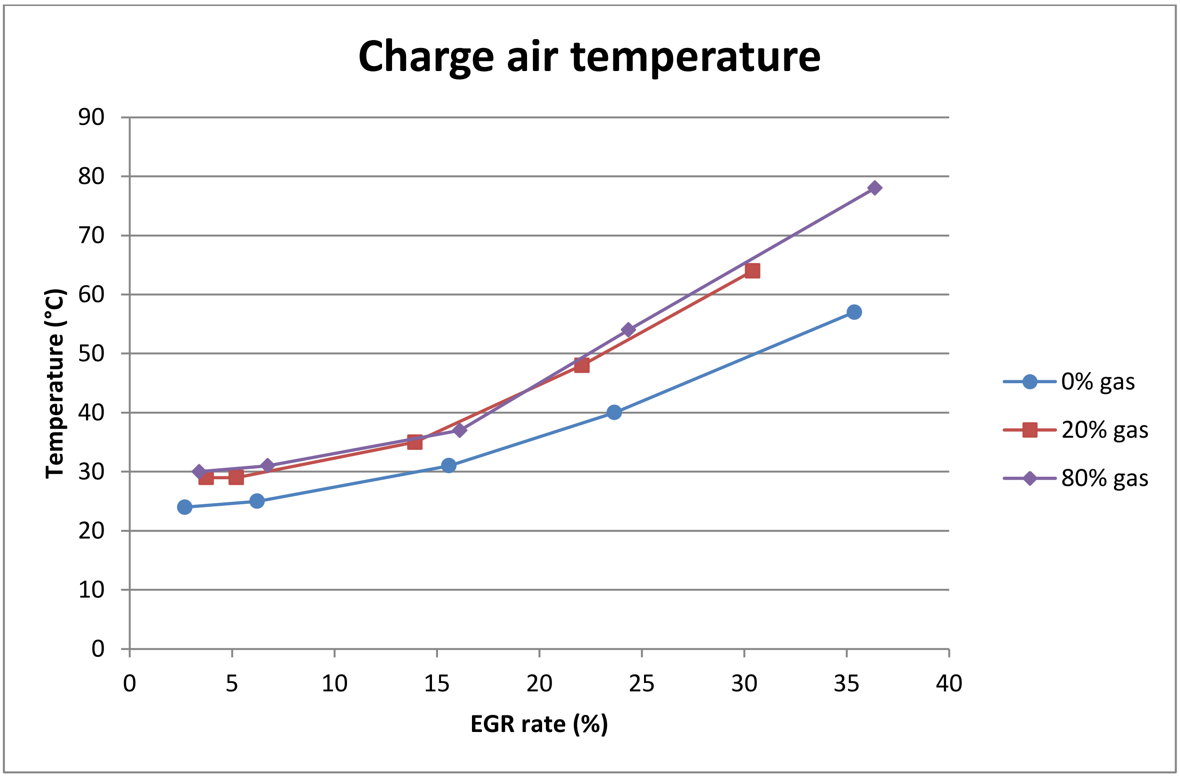

Figure 4.14: Charge air temperatures as function of EGR rate at 50% load for different gas percentages.

Increase of charge air temperatures with the EGR rate seems to be obvious, due to the raise of temperature of the exhaust gases in each point, even with a cooling system in the recirculation, which seems to be not enough.

It is also noticeable an steeper slope of the curves for the latest EGR points, especially for higher gas percentages, which could mean one of the reasons of the sudden reduction of hydrocarbon emissions at higher CH_4 percentages, although earlier start of injection could have influenced in that trend too.

This charge air temperature raise has an important effect on emissions, as it is analyzed in next chapter.

Unlike 100% load, only NO_x and THC emissions were recorded at half load, due to the problem explained in 3.4.2 of the CO range.

4.3.2. Emissions

First interesting issue that can be perceived when changing to half load is the change in the value scales of both NO_x and THC emissions. NO_x maximum values are around the double at full load, while THC are extremely lower at that mode, not for diesel but for dual-fuel performance, as it can be confirmed from figures 4.9 and 4.16. Some phenomena observed at 50% load have been studied and explained based on the full load results.

Nitrogen oxides

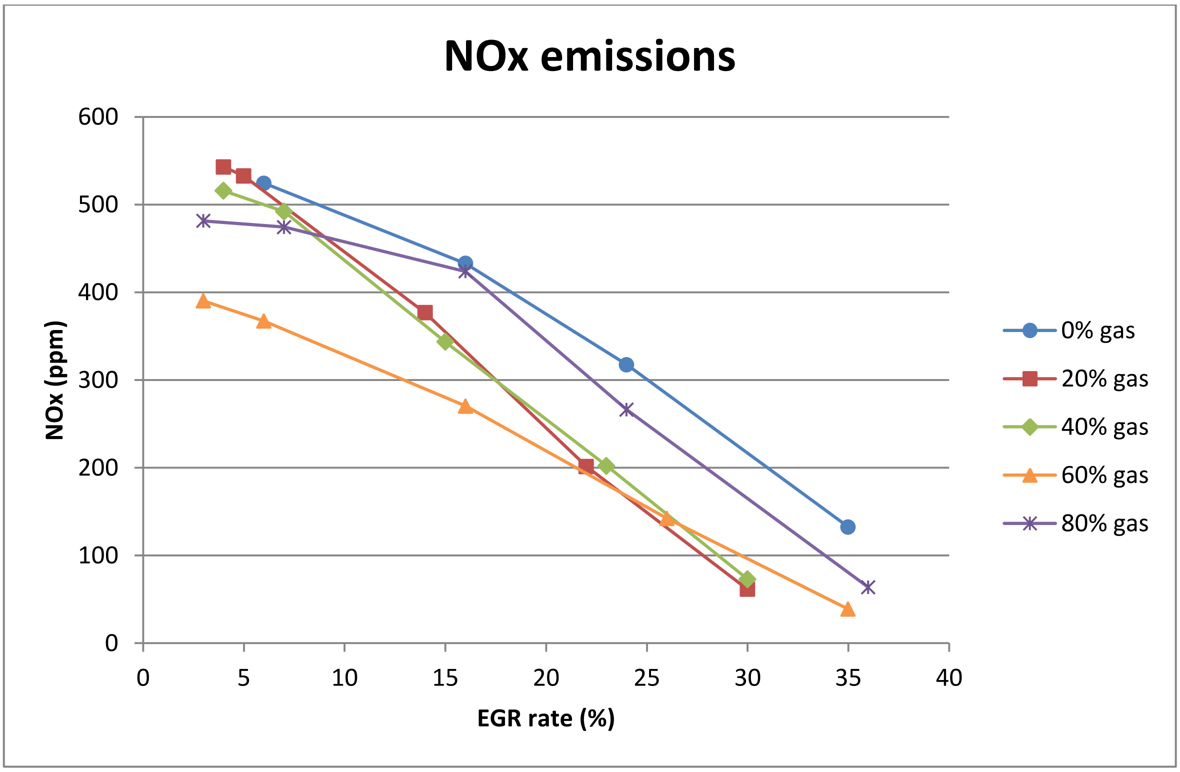

Figure 4.15: NO_x emissions as function of EGR rate at 50% load for different gas percentages.

First conclusion that can be extracted from figure 4.15 is that NO_x reductions are higher for lower gas percentages and diesel mode, as can be seen at the left of the chart, when no exhaust gas recirculation was done. Same performance was observed at full load in figure 4.8.

Curiously, 60% natural gas curve shows a notable decrease compared to the rest, but looking at figure 4.13 it can be seen that exhaust gas temperatures at these points are smaller too, even than the 80% CH_4 percentage ones. In this way, decrement in combustion temperatures seems to be evident, which implies, as observed in figure 4.15, NO_x reduction at 60% amount of natural gas.

Without taking into account gas percentage variations, evidently, NO_x emissions follow the same trend as at 100% load, decreasing while the EGR rate is increased. Effect of EGR on NO_x emissions is, thus, definitely clear for both loads.

However, in this case, NO_x are decreasing for same EGR rates when the gas quantity is raised until 60% gas, while a slightly growth can be noticed for 80% CH_4 percentage. Agarwal et al. [33], in their experiments on a single cylinder diesel engine, concluded that lower HC and significantly higher NO_x emissions were obtained with advanced injection timings. Therefore, keeping in mind that earlier SOI were used for 80% natural gas percentage, as can be seen in appendix table A1.1, it could mean an explanation for this different performance. Injection timing of 7°CA BTDC was applied during the first points, while 14°CA BTDC was employed for 80% gas test points.

Also lambda values can take part in this happening. Königsson [16] showed in his thesis that NO_x emissions decrease for lean mixtures (lower EGR rates) when the substitution rate is increased, while at lower lambdas (higher EGR rates), the behavior is entirely the opposite, increased diesel substitution rate instead leads to more NO_x.

Total unburned hydrocarbons

THC emissions performance as function of EGR rate while increasing CH_4 amounts is shown in figure 4.16.

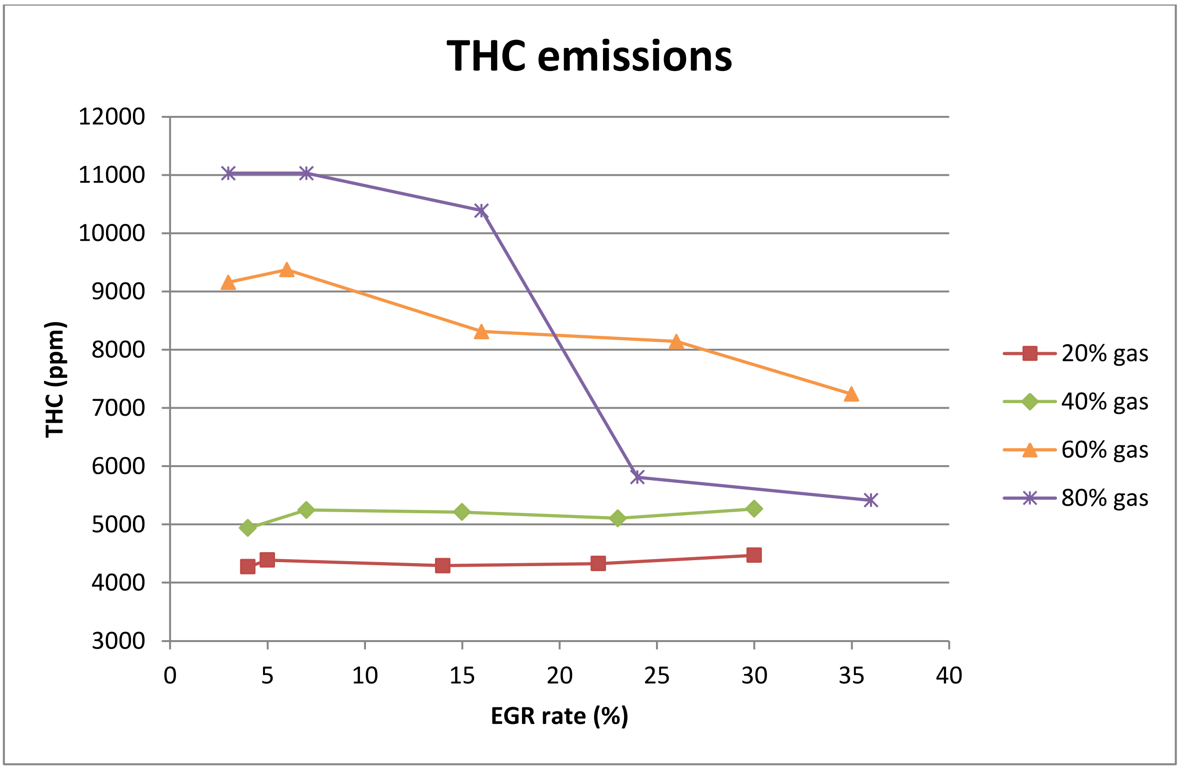

Figure 4.16: THC emissions as function of EGR rate at 50% load for different gas percentages.

As well as for NO_x emissions there is a decrease with dual-fuel mode, unburned hydrocarbon emissions go in the opposite direction. THC emissions in the experiment for diesel mode were in the range of 20-60 ppm, absolutely outside of the one in figure 4.16 (3000-12000 ppm), thus this curve has been removed for a better understanding. So generally, THC emissions have increased linearly with dual-fuel mode and no EGR applied, as it was supposed to be, especially at lower loads.

Sudden decline of THC emissions with the EGR for higher gas percentage is very clear, and it could be explained through the methane lambda. This lambda is approaching to stoichiometric while increasing the EGR rate, so this can mean a better mixture and less methane slip.

When decreasing the engine load, there is a sharp increase of THC emissions range under dual-fuel mode (200-1400 ppm at 100% load versus 5000-10000 ppm at 50% load). This can be the result of the increase in combustion temperature that helps to oxidize efficiently the hydrocarbons at full load.

As said, without EGR rate variation, THC emissions grow while increasing methane amount as they are supposed to. Differences start when modifying EGR rate. THC keep practically constant for lower natural gas percentages, however different performances can be observed for the highest. As commented at 100% load, a growing reduction in hydrocarbons formation is noticed when increasing the gas amount, so same explanation should be useful in this case. Higher THC emissions without EGR could be explained by excessive amounts of methane slip, since it is probable that a large separation from the stoichiometric lambda implies incomplete methane burning, since leaner mixtures were used without EGR.

Finally, trying to find an explanation of the sudden decline of THC emissions when increasing the EGR rate with 80% gas, Agarwal et al. [33] conclusions can also help. As SOI was incremented in this case to 14°CA BTDC, it could imply a reduction in THC emissions. However, it should happen to the whole 80% gas curve, and it does not. An interpretation of this could be found in the higher amounts of methane slip with lower EGR rates, as commented above.

On the other hand, charge air temperatures raise with EGR could also intervene. As observed in figure 4.14, charge air temperature has a steeper growth for higher gas percentages. So that this more intense decrease of THC could also be due to combustion reactions improvement and faster combustion when extremely high charge air temperatures appear [34].

5. Conclusion

The main research objective in this master's thesis was to study the exhaust gas recirculation effect on NO_x, THC and CO emissions on dual-fuel methane combustion in a single cylinder high-speed research engine. In order to explain this effect, engine performance was also analyzed considering different parameters, such as lambdas, temperatures, cylinder pressures, HRR or HR.

Background of diesel engines shows that EGR cause a clear decrement on NO_x emissions, while increasing THC and CO levels, as well as an increase on soot emissions due to the worst diffusion combustion and soot oxidation. However, not many studies have been done to examine EGR parameter effect on dual-fuel engines, especially because also other parameters are usually modified at the same time. Because of this reason, there is not a clear conclusion on how EGR variation affects THC and CO emissions. Most of the parameters were kept constant in this thesis, with the only variation of the gas-diesel amounts, to study the dual-fuel performance, and the EGR rate. In case some other parameter, such as SOI, was modified it is clearly specified in appendix A1 and commented in the test program, chapter 4.1. Also charge air temperatures were increasing even with an EGR cooler before the intake manifold, but it was concluded that this raise did not have an important attenuation on the EGR effect, since increase of charge air temperature generally leads to higher NO_x.

Maximum methane percentages used in the thesis were previously researched by the Department of Energy Technology of Aalto University, based on different experiments. Maximum cylinder peak pressure was the restricting factor especially at full load, so pressures and SOI were modified in order not to overcome peak pressures around 240 bar, since engine design would not resist it.

Both CH_4 percentage and EGR rate increments influence on emissions has been analysed, as well as their combination. Different performance was found in some cases when increasing the amount of natural gas, depending on the EGR rate. That is to say, different behaviors were found between lower and higher EGR rates with the increase of the methane amount. For a better understanding, a small summary of the results has been done:

- NO_x emissions mainly decrease when changing to dual-fuel mode, as well as there is an important reduction with the EGR rate. Moreover, the effect of EGR is higher for lower natural gas percentages.

- THC emissions clearly increase with the CH_4 percentage. The effect of EGR basically depends on the amount of gas injected. At half load, EGR starts to cause a reduction of THC emissions from 60% methane percentage, and this effect is even higher for highest natural gas amounts. At full load, THC have a small raise in diesel mode due to the EGR, while the reduction is immediate when changing to dual-fuel mode.

- CO emissions undergo an increment when changing to dual-fuel mode. Nevertheless, no methane – gas ratio variation effect on CO can be clearly established. CO emissions keep practically constant with the EGR rate, but a sudden and high increase appears for the highest EGR rates, approximately when they exceed 20%.

In terms of engine performance, EGR rate increment mainly causes a decrease in the lambda, due to the oxygen concentration reduction. The combustion seems to be less efficient and faster, and ignitions delays are lower for the highest EGR rates, even though all these tendencies are not linear with the EGR rate increase. Evidently, EGR also causes an increase in exhaust gas temperatures, and even in charge air temperatures, so the EGR cooling system used in this research engine was not enough to maintain intake temperatures around the same values.

To summarize, in dual-fuel combustion NO_x emissions are lower, while THC and CO are higher than diesel. EGR causes a clearly decrease on NO_x emissions, reducing also THC, especially at higher amounts of methane. However, EGR effect on CO emissions is low until EGR rates are around 20%, moment in which there is a large increase.

6. Future work

Tests made in this master's thesis will provide the research group of Internal Combustion Engines of Aalto University a good base in order to improve their dual-fuel experiments. Next step would be to confirm the results obtained here, taking into consideration all the values achieved in this work. The most important principle when drawing conclusions in research works is to be certain that the quality of the measurements is accurate enough to formulate a scientific conclusion.

Optical research studies could also help to analyse and confirm performances found in these measurements, as they supply images of what happens inside the combustion chamber. It is a very helpful tool, which could show the evolution of the combustion and, thus, provide some explanations on the emission formation and mixture combustion. It would be especially interesting since dual-fuel combustion is more complex than gasoline and diesel ones, and it would be useful to know how both diesel and methane burn inside the cylinder in order to understand the engine performance.

Regarding to the engine design, some variations could be made as improvements. A throttled version of the engine would be useful for a better control of the lambda, since EGR implies high changes in this parameter. It would allow a better trade-off between NO_x, THC and CO emissions.

Lastly, also a better EGR cooler would help to supply a complete EGR cooling. Charge air temperatures in this thesis were increasing as the EGR rate was raised, occurrence that should not happen when cooling the exhaust gases. Therefore, an improvement in the EGR cooling system that would be able to keep charge air temperatures constant should be considered in future research works with this engine.

Of course, thinking about improvements, the control system should be also implemented or changed by a new one, in order to avoid all the problems found during the realization of this thesis at the time of taking measurements.

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8. Appendices

A1. Engine measurement matrices

A1.1. Measurement matrix for 50% load

Point	%Gas	Gas injection duration (ms)	Diesel injection duration (µs)	SOI	Charge air pressure (bar)	Exhaust gas pressure (bar)
1	0	0	834	-7	1.40	1.55
2	0	0	834	-7	1.40	1.55
3	0	0	834	-7	1.40	1.55
4	0	0	834	-7	1.40	1.55
5	0	0	834	-7	1.40	1.55
6	20	5.2	720	-7	1.40	1.55
7	20	5.2	720	-7	1.40	1.55
8	20	5.2	720	-7	1.40	1.55
9	20	5.2	720	-7	1.40	1.55
10	20	5.2	720	-7	1.40	1.55
11	40	5.8	670	-7	1.40	1.55
12	40	5.8	670	-7	1.40	1.55
13	40	5.8	670	-7	1.40	1.55
14	40	5.8	670	-7	1.40	1.55
15	40	5.8	670	-7	1.40	1.55
16	60	7.1	570	-7	1.40	1.55
17	60	7.1	570	-7	1.40	1.55
18	60	7.1	570	-7	1.40	1.55
19	60	7.1	570	-7	1.40	1.55
20	60	7.1	570	-7	1.40	1.55
21	80	8.4	435	-14	1.40	1.55
22	80	8.4	435	-14	1.40	1.55
23	80	8.4	435	-14	1.40	1.55
24	80	8.4	435	-14	1.40	1.55
25	80	8.4	435	-14	1.40	1.55
26	90	9.2	365	-18	1.40	1.55
27	90	9.2	365	-18	1.40	1.55
28	90	9.2	365	-18	1.40	1.55
29	90	9.2	365	-18	1.40	1.55
30	90	9.2	365	-18	1.40	1.55

A1.2. Temperatures matrix for 50% load

Point	%Gas	Charge air temperature (°C)	Exhaust gas temperature (°C)
1	0	24	270
2	0	25	275
3	0	31	275
4	0	40	285
5	0	57	295
6	20	29	360
7	20	29	364
8	20	35	372
9	20	48	390
10	20	64	415
11	40	30	362
12	40	31	366
13	40	37	376
14	40	45	385
15	40	57	410
16	60	25	325
17	60	28	340
18	60	36	365
19	60	54	400
20	60	76	435
21	80	30	285
22	80	31	290
23	80	37	335
24	80	54	370
25	80	78	410
26	90	27	340
27	90	28	355
28	90	42	395
29	90	55	420
30	90	78	440

A1.3. EGR rate measurement matrix for 50% load

Point	%Gas	Intake CO_2 (%vol)	Exhaust CO_2 (%vol)	EGR valve (%)	EGR rate (%)
1	0	0.1	3.7	0	3
2	0	0.24	3.85	5	6
3	0	0.68	4.36	10	16
4	0	1.25	5.28	15	24
5	0	2.3	6.5	20	35
6	20	0.18	4.8	0	4
7	20	0.25	4.8	5	5
8	20	0.78	5.6	10	14
9	20	1.48	6.7	15	22
10	20	2.54	8.35	20	30
11	40	0.18	4.76	0	4
12	40	0.33	4.83	5	7
13	40	0.8	5.5	10	15
14	40	1.45	6.34	15	23
15	40	2.47	8.13	20	30
16	60	0.13	3.88	0	3
17	60	0.26	4.12	5	6
18	60	0.79	5.05	10	16
19	60	1.65	6.4	15	26
20	60	3.12	8.9	20	35
21	80	0.11	3.23	0	3
22	80	0.23	3.41	5	7
23	80	0.75	4.65	10	16
24	80	1.5	6.16	15	24
25	80	3.06	8.41	20	36
26	90	0.13	4.05	0	3
27	90	0.26	4.28	5	6
28	90	0.98	5.95	10	16
29	90	2.26	7.81	15	29
30	90	3.57	9.5	20	38

A1.4. Measurement matrix for 100% load

Point	%Gas	Gas injection duration (ms)	Diesel injection duration (µs)	SOI	Charge air pressure (bar)	Exhaust gas pressure (bar)
31	0	0	2200	-5	2.55	2.70
32	0	0	2200	-5	2.55	2.70
33	0	0	2200	-5	2.55	2.70
34	0	0	2200	-5	2.55	2.70
35	0	0	2200	-5	2.55	2.70
36	0	0	2200	-5	2.55	2.70
37	20	6.4	1200	-5	2.55	2.70
38	20	6.4	1200	-5	2.55	2.70
39	20	6.4	1200	-5	2.55	2.70
40	20	6.4	1200	-5	2.55	2.70
41	20	6.4	1200	-5	2.55	2.70
42	40	8	620	-5	2.55	2.70
43	40	8	620	-5	2.55	2.70
44	40	8	620	-5	2.55	2.70
45	40	8	620	-5	2.55	2.70
46	40	8	620	-5	2.55	2.70
47	40	8	620	-5	2.55	2.70

A1.5. Temperatures matrix for 100% load

Point	%Gas	Charge air temperature (°C)	Exhaust gas temperature (°C)
31	0	26	475
32	0	29	480
33	0	34	482
34	0	42	495
35	0	55	515
36	0	65	540
37	20	27	510
38	20	31	510
39	20	39	530
40	20	60	565
41	20	85	595
42	40	28	460
43	40	31	460
44	40	38	460
45	40	50	488
46	40	68	520
47	40	90	540

A1.6. EGR rate measurement matrix for 100% load

Point	%Gas	Intake CO_2 (%vol)	Exhaust CO_2 (%vol)	EGR valve (%)	EGR rate (%)
31	0	0.17	4.88	0	3
32	0	0.29	4.85	5	6
33	0	0.59	5.31	10	11
34	0	1.01	5.95	15	17
35	0	1.65	6.97	20	24
36	0	2.40	8.00	25	30
37	20	0.22	7.32	0	3
38	20	0.39	7.49	5	5
39	20	0.84	8.38	10	10
40	20	1.55	9.60	15	16
41	20	2.75	11.54	20	24
42	40	0.24	6.45	0	3
43	40	0.40	6.62	5	6
44	40	0.68	7.00	10	10
45	40	1.40	8.15	15	17
46	40	2.65	9.80	20	24
47	40	3.25	11.18	25	29

A2. Emission measurement matrices

A2.1. Emissions matrix for 50% load

Point	%Gas	O_2 (%vol)	CO_2 (%vol)	CO (ppm)	THC (ppm)	NO_x (ppm)	NO_2 (ppm)	NO (ppm)
1	0	15.49	3.86	10.38	20.73	524.23	20.11	500.02
2	0	14.72	4.44	17.47	29.64	432.94	19.79	410.3
3	0	13.7	5.19	18.55	38.22	317.39	15.84	300.58
4	0	12.11	6.37	29.09	60.33	132.53	11.06	116.75
6	20	13.33	4.82	582.64	4272.96	542.85	163.55	350.43
7	20	13.34	4.78	582.8	4388.97	532.42	186.39	310.36
8	20	12.04	5.65	582.72	4293.39	376.75	149.52	200.27
9	20	10.61	6.57	582.83	4327.23	201.44	91.7	96.21
10	20	7.71	8.48	582.35	4470.09	61.51	27.4	27.74
11	40	13.22	4.77	582.76	4938.66	515.69	167.65	318.05
12	40	13.15	4.83	582.82	5246.94	491.89	178.41	283.55
13	40	12.05	5.52	582.48	5212.02	343.47	139.55	179.14
14	40	10.76	6.38	583.26	5105.25	202.09	91.79	96.48
15	40	7.76	8.27	582.58	5266.32	73.06	34.53	33.2
16	60	14.19	3.89	582.98	9158.82	390.11	159.79	198.17
17	60	13.84	4.12	582.97	9373.2	367.3	160.8	178.81
18	60	12.34	5.07	582.65	8312.91	270.37	121.05	127.71
19	60	10.12	6.41	582.75	8141.1	142.38	69.91	59.76
20	60	5.96	8.91	582.76	7238.4	38.84	13.69	22.16
21	80	14.93	3.22	582.84	11031.36	481.44	107.36	358.6
22	80	14.62	3.42	582.84	11031.36	474.26	112.44	340.69
23	80	12.33	4.79	582.88	10386.93	423.78	96.14	312.14
24	80	10.07	6.17	582.79	5810.1	266.2	54.45	200.79
25	80	6.21	8.43	582.68	5415.12	63.68	16.27	45.32
26	90	13.46	4.03	582.86	9314.55	456.28	68.88	375.69
27	90	13.04	4.25	582.74	8380.62	398.88	53.42	336.79
28	90	10.20	6	582.68	3394.89	368.35	40.25	321.96
29	90	7.25	7.76	582.87	3123.48	197.74	20.7	172.26
30	90	3.33	9.68	582.59	7966.68	24.34	7.36	15.32

A2.1. Emissions matrix for 100% load

Point	%Gas	O_2 (%vol)	CO_2 (%vol)	CO (ppm)	THC (ppm)	NO_x (ppm)	NO_2 (ppm)	NO (ppm)
31	0	5.38	4.87	69.35	51.96	1222.45	50.11	973.67
32	0	5.25	4.95	121.08	69.33	1103.21	52.36	904.83
33	0	4.75	5.3	136.25	70.05	858.02	44.59	709.96
34	0	3.83	5.96	132.71	73.08	582	35.25	546.45
35	0	2.40	7.01	130.26	76.62	328.99	22.07	307.58
36	0	0.96	8.06	170.88	76.98	175.4	12.35	164.94
37	20	9.86	7.32	485.94	871.26	1194.03	107.19	1071.13
38	20	9.61	7.49	572.8	907.44	1011.73	103.99	892.37
39	20	8.32	8.38	488.35	851.82	708.6	72.97	620.81
40	20	6.48	9.59	528.86	795.87	334.29	49.47	276.15
41	20	3.41	11.45	2620.23	602.16	97.17	34.36	57.65
42	40	10.66	6.44	434.3	1317.51	1144.69	104.08	1028.05
43	40	10.40	6.61	417.32	1273.32	1034.86	94.31	930.58
44	40	9.76	7	410.91	1263.84	830.62	78.28	742.67
45	40	7.95	8.16	406.08	1197.78	474.8	47.07	421.5
46	40	5.34	9.79	728.3	1144.23	152.34	26.85	123.33
47	40	2.86	11.17	2851.24	863.64	51.7	11.42	37.73

A3. Data processing Matlab scripts

A3.1. Script for extracting measurement data and plotting curves

```
% This script extracts measurements made with LEO Research Engine
% It calculates and plots Cylinder Pressures, Heat Release rates and Cumulative Heat
Release rates

files = dir('*.mat');
n = size(files(:,1),1);

q = 1; % File number of measurement data


% Extracting measurement data:
for i=1:n
load(files(i,1).name);

for t=1:26
measurements = struct2cell(load(files(i,1).name));
var = cell2mat(measurements);
data = struct2cell(var);
var2 = data(2,1);
var3 = cell2mat(var2);
Label{t,1} = var3(1,t).Name;
DATA{t,1} = var3(1,t).Data';

end


% arranging data
angle(:,i) = cell2mat(DATA(1,:));
Angle(:,1) = angle(:,i);
chargeFlow(:,i) = cell2mat(DATA(2,:));
averageChargeFlow(:,i) = mean(chargeFlow(:,i));
chargePress(:,i) = cell2mat(DATA(3,:));
p_intake(:,1) = chargePress(:,i);
averageChargePress(:,i) = mean(chargePress(:,i));
offset(:,1) = averageChargePress(:,i);
chargeTemp(:,i) = cell2mat(DATA(5,:));
averageChargeAirTemp(:,i) = mean(chargeTemp(:,i));
cylPress(:,i) = cell2mat(DATA(6,:));
cyl_P(:,1) = cylPress(:,i);
cylPressFirstCycle(:,i) = cylPress(1:1000,i);
fuelMassFlow(:,i) = cell2mat(DATA(11,:));
averageFuelMassFlow(:,i) = mean(fuelMassFlow(:,i));
CH4MassFlow(:,i) = cell2mat(DATA(26,:));
averageCH4MassFlow(:,i) = mean(CH4MassFlow(:,i));
relHumidity(:,i) = cell2mat(DATA(17,:));
averageRelHumidity(:,i) = mean(relHumidity(:,i));

start_HR_ca = 1
end_HR_ca = 1000
start_cylp_ca = 1
end_cylp_ca = 1003


heatrelease_leo_2(Angle, cyl_P, p_intake, offset); % heat release calculation


% saving the variables
dQ(i,:) = hr(:,:);
cylPress_hr(i,:) = P_cyl_hr(1,:)/100000;
intake_p_hr(i,:) = P_intake_hr/100000;
Angle_hr(i,:) = angle_hr(1,:);
cumQ(i,:) = cumsum(hr(:,:));

end
```

```
%% Plots

% Plotting Heat Release Rate Curves (HRR)
figure();

hold all

plot(Angle_hr(1,:), dQ(1,:), 'b', 'LineWidth',1.5);
plot(Angle_hr(6,:), dQ(6,:), 'k', 'LineWidth',1.5);
plot(Angle_hr(2,:), dQ(2,:), 'r', 'LineWidth',1.5);
plot(Angle_hr(3,:), dQ(3,:), 'g', 'LineWidth',1.5);
plot(Angle_hr(4,:), dQ(4,:), 'c', 'LineWidth',1.5);
plot(Angle_hr(5,:), dQ(5,:), 'm', 'LineWidth',1.5);

h_legend = legend('3% EGR', '6% EGR', '10% EGR', '17%EGR', '24% EGR', '29%EGR');
set(h_legend,'FontSize',25);
title('Heat release rate','FontSize',35);
xlabel('Crank angle (deg)','FontSize',20);
ylabel('HRR (J/deg)','FontSize',20);
grid on;

xlim(gca, [-20 60]);
ylim(gca, [-20 500]);


% Plotting Cylinder Pressure Curves
figure();

hold all

plot(Angle_hr(1,:), cylPress_hr(1,:), 'b', 'LineWidth',1.5);
plot(Angle_hr(6,:), cylPress_hr(6,:), 'k', 'LineWidth',1.5);
plot(Angle_hr(2,:), cylPress_hr(2,:), 'r', 'LineWidth',1.5);
plot(Angle_hr(3,:), cylPress_hr(3,:), 'g', 'LineWidth',1.5);
plot(Angle_hr(4,:), cylPress_hr(4,:), 'c', 'LineWidth',1.5);
plot(Angle_hr(5,:), cylPress_hr(5,:), 'm', 'LineWidth',1.5);

h_legend = legend('3% EGR', '6% EGR', '10% EGR', '17%EGR', '24% EGR', '29%EGR');
set(h_legend,'FontSize',25);
title('Cylinder pressure','FontSize',35);
xlabel('Crank angle (deg)','FontSize',20);
ylabel('Pressure (bar)','FontSize',20);
grid on;

xlim(gca, [-40 60]);
ylim(gca, [50 250]);


% Plotting Cumulative Heat Release Rate Curves (HR)
figure();

hold all

plot(Angle_hr(1,:), cumQ(1,:), 'b', 'LineWidth',1.5);
plot(Angle_hr(6,:), cumQ(6,:), 'k', 'LineWidth',1.5);
plot(Angle_hr(2,:), cumQ(2,:), 'r', 'LineWidth',1.5);
plot(Angle_hr(3,:), cumQ(3,:), 'g', 'LineWidth',1.5);
plot(Angle_hr(4,:), cumQ(4,:), 'c', 'LineWidth',1.5);
plot(Angle_hr(5,:), cumQ(5,:), 'm', 'LineWidth',1.5);

h_legend = legend('3% EGR', '6% EGR', '10% EGR', '17%EGR', '24% EGR', '29%EGR');
set(h_legend,'FontSize',25);
title('Cumulative heat release rate','FontSize',35)
xlabel('Crank angle (deg)','FontSize',20)
ylabel('CHRR (J)','FontSize',20)

grid on
xlim(gca, [-20 60]);
```

A3.2. Script for cylinder pressure and heat release calculation

```
% This script performs cylinder pressure and heat release calculations:

function heatrelease_leo_2(Angle,cyl_P,p_intake,offset)

conrodLength = 0.232; % Connecting rod length
crankRadius = 0.0725; % Crank radius
lambda= crankRadius/conrodLength; % Cranksaft ratio
Vd1=0.00140315; % Displacement volume
gamma=1.35; % from literature
Rc=16.5; %Compression ratio updated 09.09.2013
Vc=Vd1/(Rc-1); % Free Volume at TDC, updated 09.09.2013
D=0.1108; % Piston Diameter


f=36000; % sampling frequency
f_cutoff = 2000; % cutoff frequency
fnorm = f_cutoff/(f/2); % normalized cut off freq, you can change it to any value depending
on your requirements
[b1,a1] = butter(10,fnorm,'low'); % Low pass Butterworth filter of order 10
p_cyl = filtfilt(b1,a1,cyl_P); % filtering


assignin('base','P_CYL',p_cyl);
angle_hr = Angle;
assignin('base','Angle360360',angle_hr);


%% re-arregement of the measurement data:
y = 0;
x = 0;
intake = 0;
cycle = 1;
e = 2;

if length(p_cyl) == length(angle_hr);
    len = length(p_cyl);
    y(1,1) = p_cyl(1);
    intake(1,1) = p_intake(1);
    x(1,1) = angle_hr(1);

    for r=2:len

        if (angle_hr(r-1) > 0 && angle_hr(r) < 0)
            cycle = cycle + 1;
            e = 1;
        end
        y(cycle,e) = p_cyl(r);
        intake(cycle,e) = p_intake(r);
        x(cycle,e) = angle_hr(r);
        e = e+1;
    end
end

a = x(2:25, :); % removing the first and last row
b = y(2:25, :);
c = intake(2:25, :);


% calculate averages for cylinder and intake pressure
avg_p_cyl = mean(b)*100000;
avg_angle = mean(a);
avg_intake_p = mean(c)*100000;


% Decrease all cylinder pressure measurements to the same length
for r=1:1000
Avg_p_cyl(1,r) = avg_p_cyl(1,r);
Avg_angle(1,r) = avg_angle(1,r);
Avg_intake_p(1,r) = avg_intake_p(1,r);
```

```
end


% set the pressure level
Avg_p_cyl = Avg_p_cyl+(mean(Avg_intake_p(1:200))-mean(Avg_p_cyl(1:200)));
Avg_p_cyl = Avg_p_cyl+offset;


%calculate radian angles for next step
x_rad = Avg_angle/360*2*pi;

assignin('base','a',a);
assignin('base','b',b);
assignin('base','c',c);


% Output the averaged and filtered cylinder pressure to workspace
assignin('base','P_cyl_hr',Avg_p_cyl);
assignin('base','P_intake_hr',Avg_intake_p);
assignin('base','angle_hr',Avg_angle);
dQ = zeros(1,length(x_rad));
dV = Vd1/2.*(sin(x_rad)+lambda.*sin(x_rad).*cos(x_rad)./sqrt(1-lambda^2.*sin(x_rad).^2));
for s = 2:(length(x_rad)-1)
dQ(1,s) = (gamma/(gamma-1)*Avg_p_cyl(s)*dV(s)+1/(gamma-1)*...
(Vc+pi*D*D/4*(conrodLength+crankRadius-
(crankRadius*cos(x_rad(s))+sqrt(conrodLength*conrodLength-...
crankRadius*crankRadius*sin(x_rad(s))*sin(x_rad(s))))))*(Avg_p_cyl(s+1)-Avg_p_cyl(s-
1))/(x_rad(s+1)-x_rad(s-1)))/(360/(2*pi));
end


%Smoothens heat release curve:
dQ=smooth(Avg_angle,dQ);

assignin('base','hr',dQ);
```